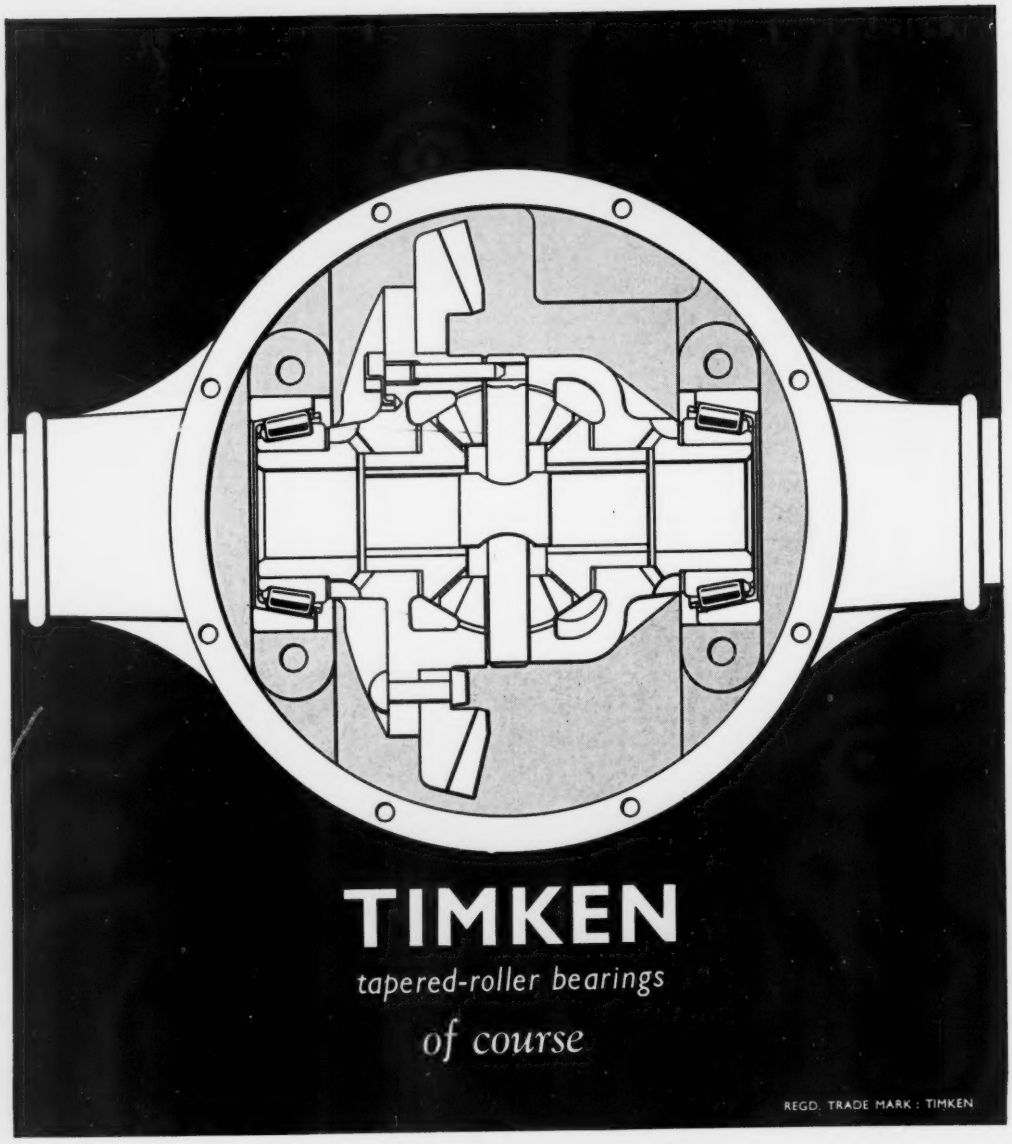
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 42 No. 558. OCTOBER, 1952 PRICE: 3s. 6d.



Jaguar XK 120

for racing

TORSION BARS

for trouble free

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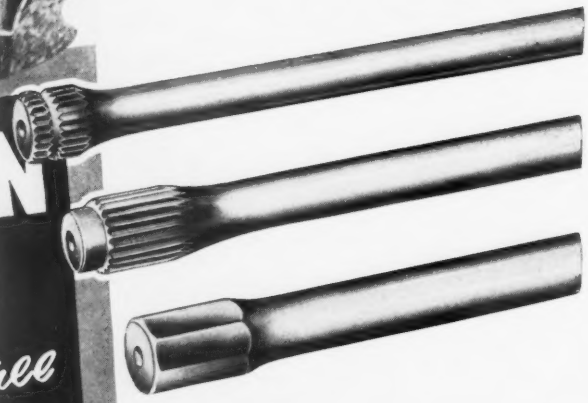

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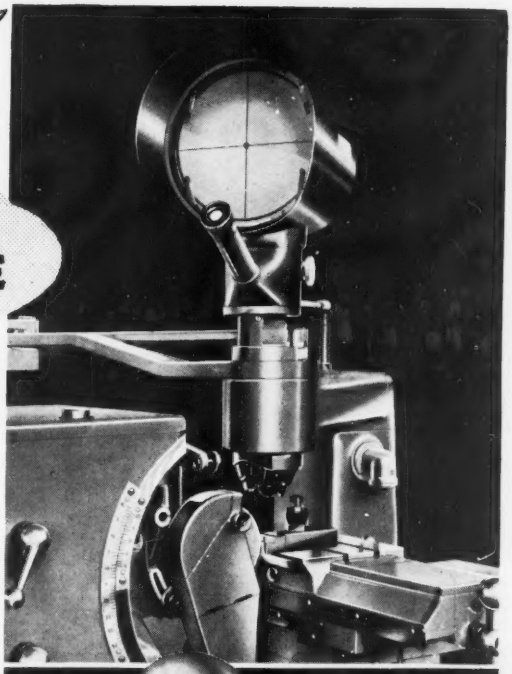

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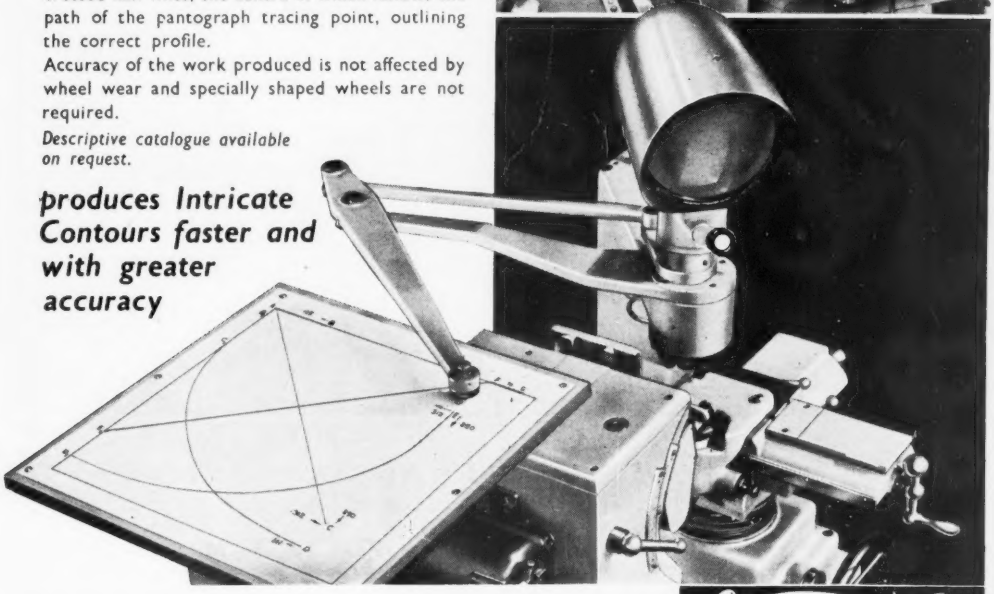

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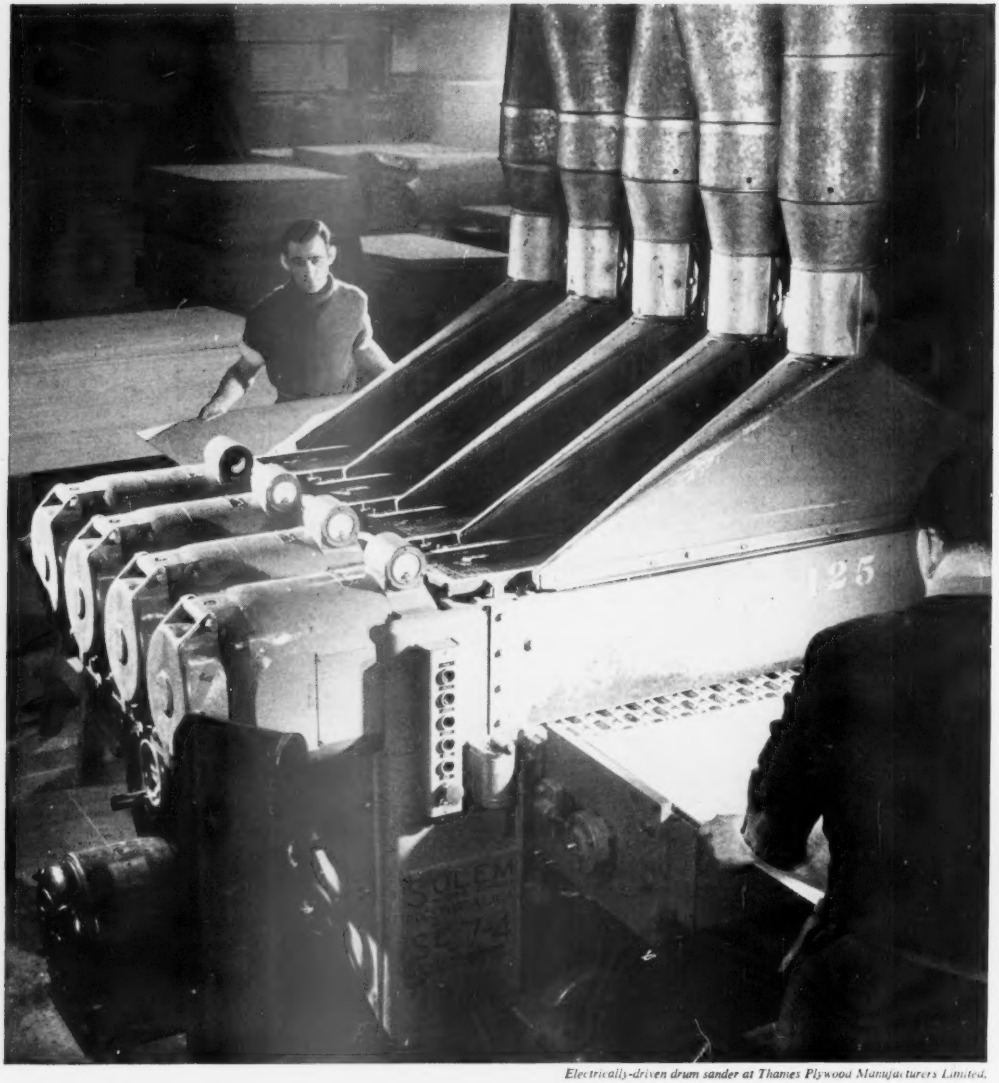

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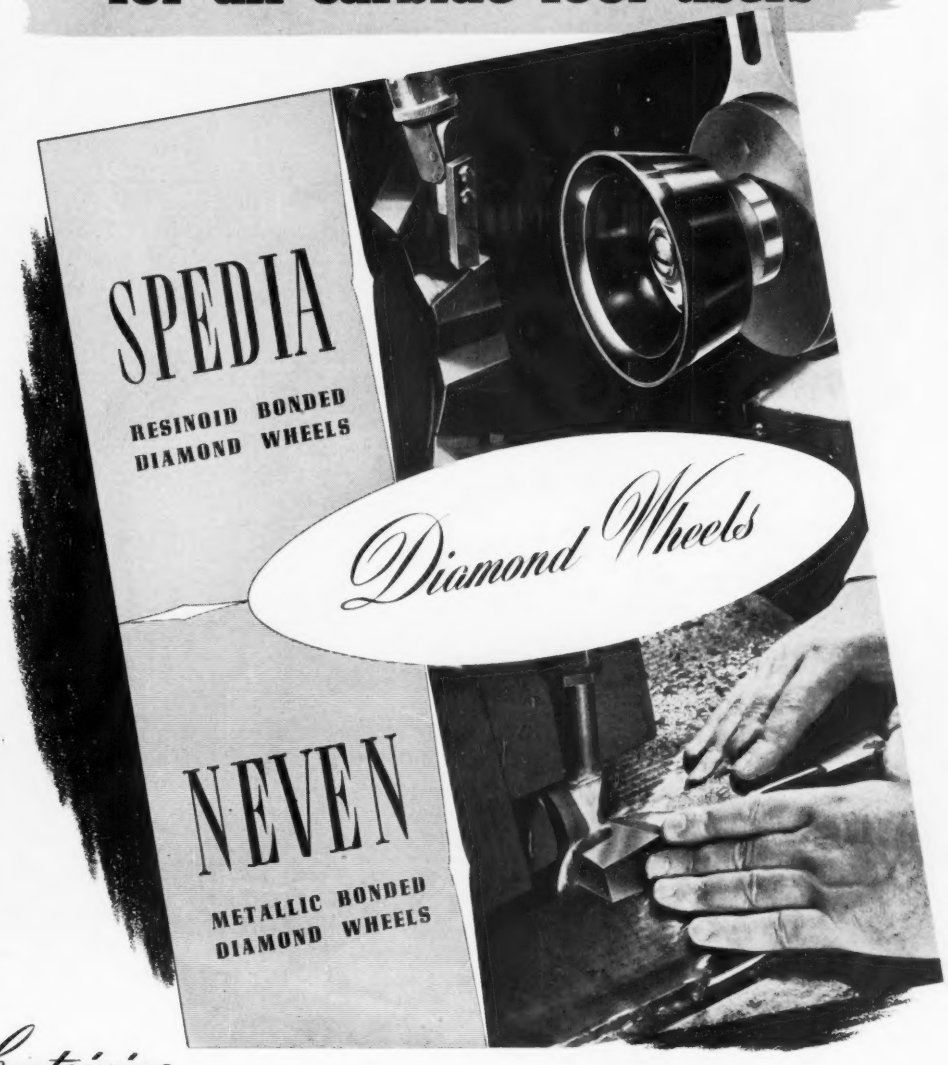

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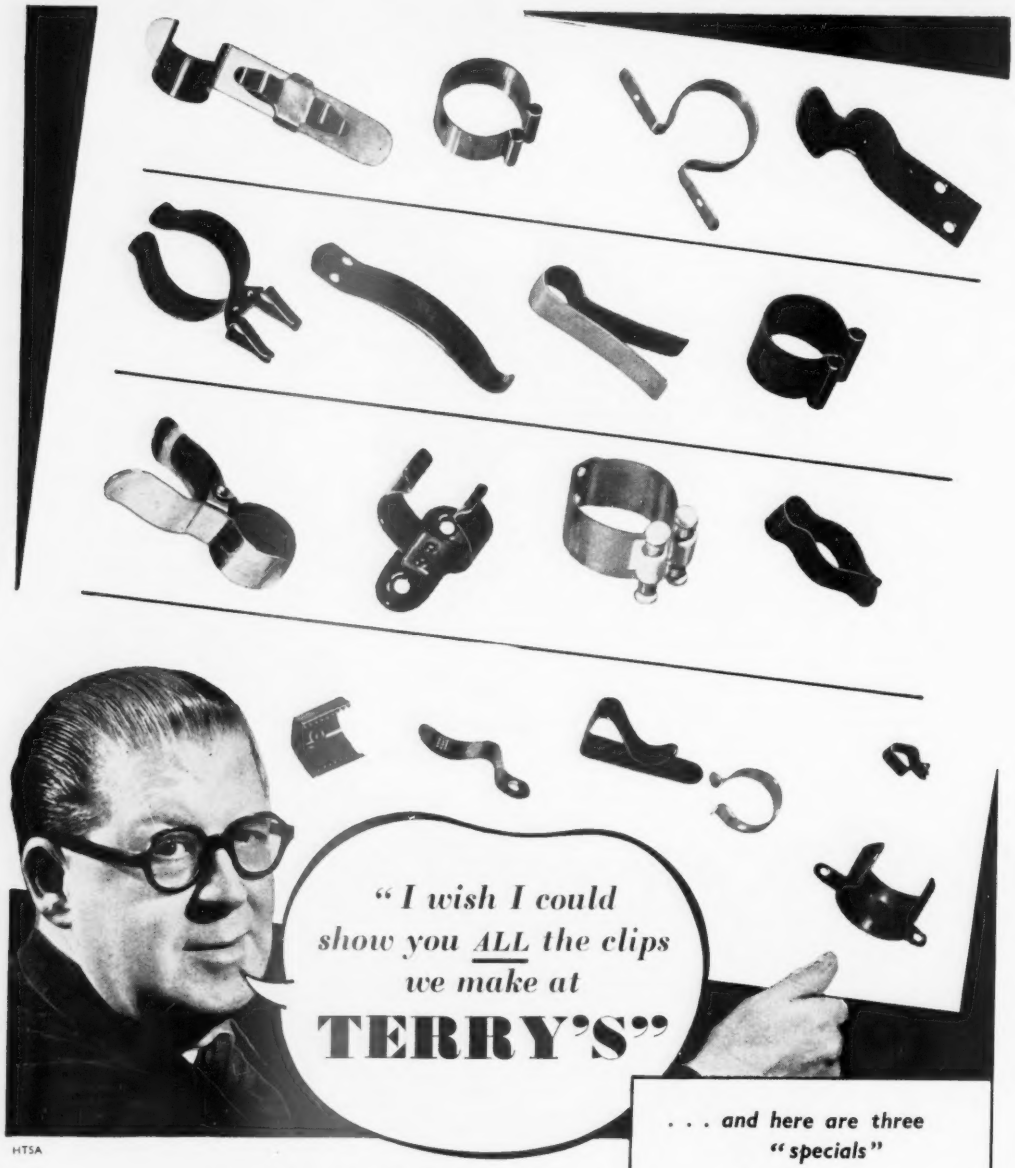

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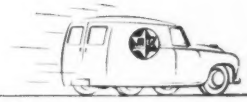

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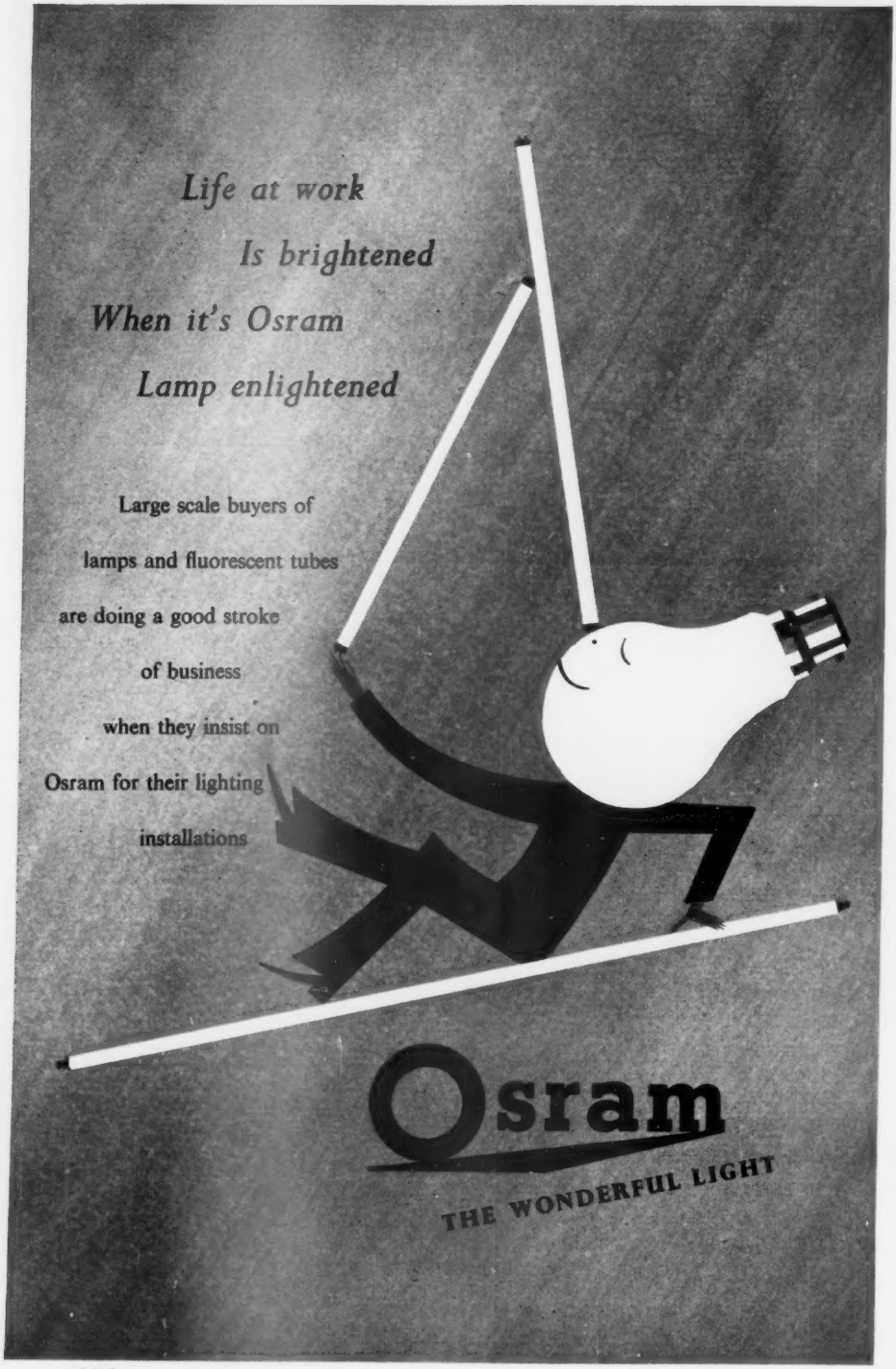

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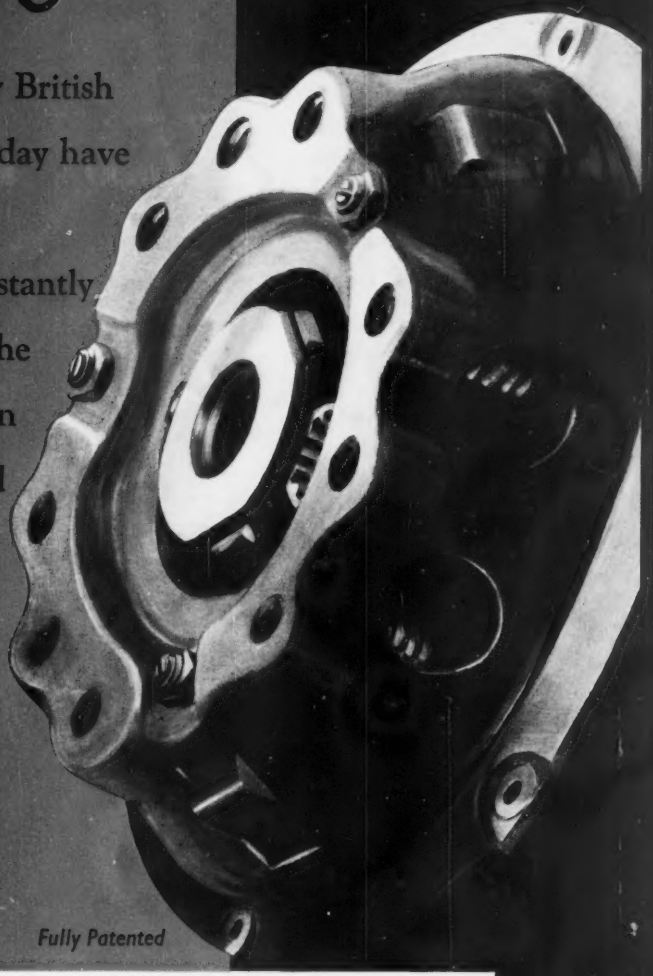

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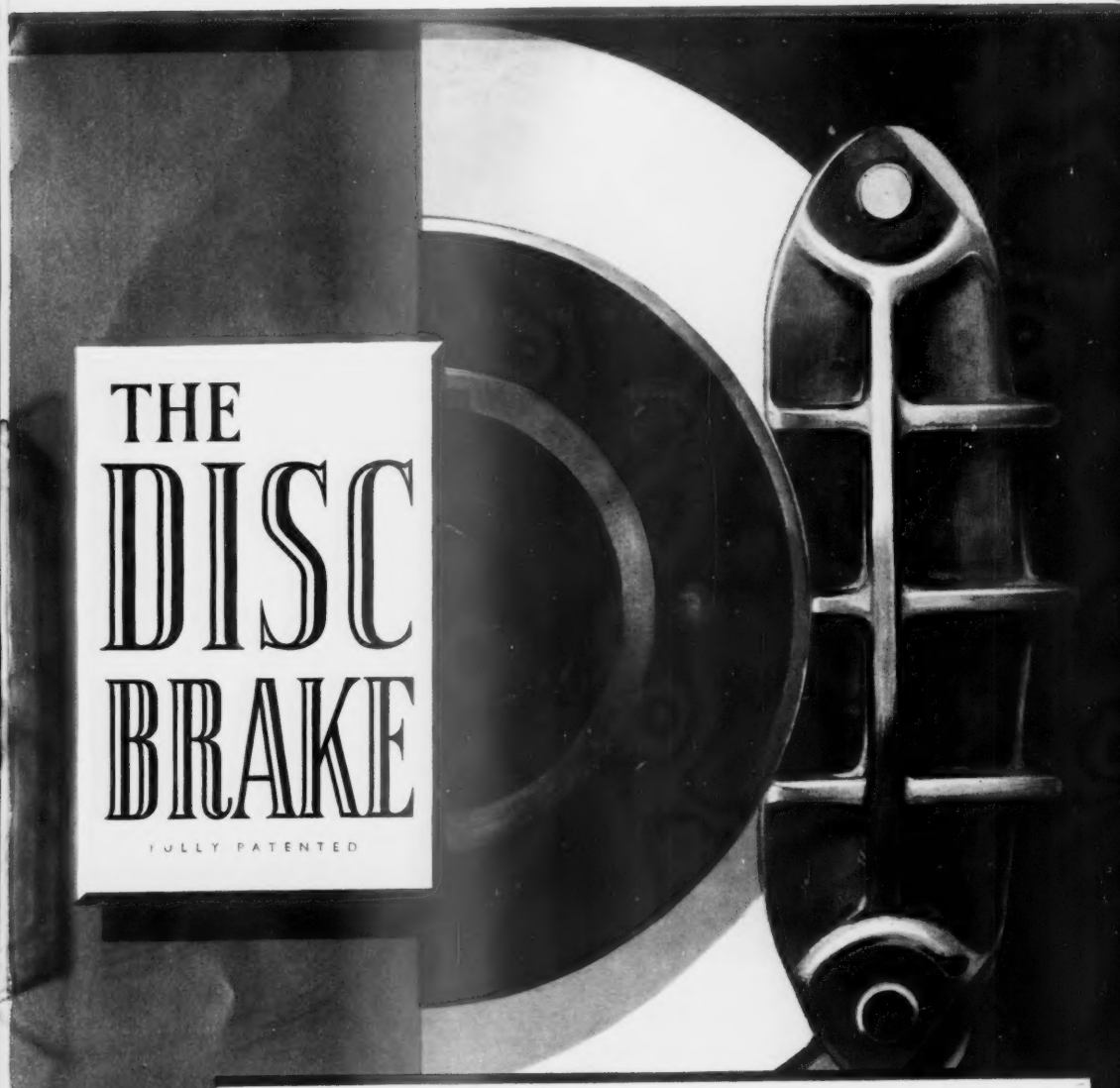

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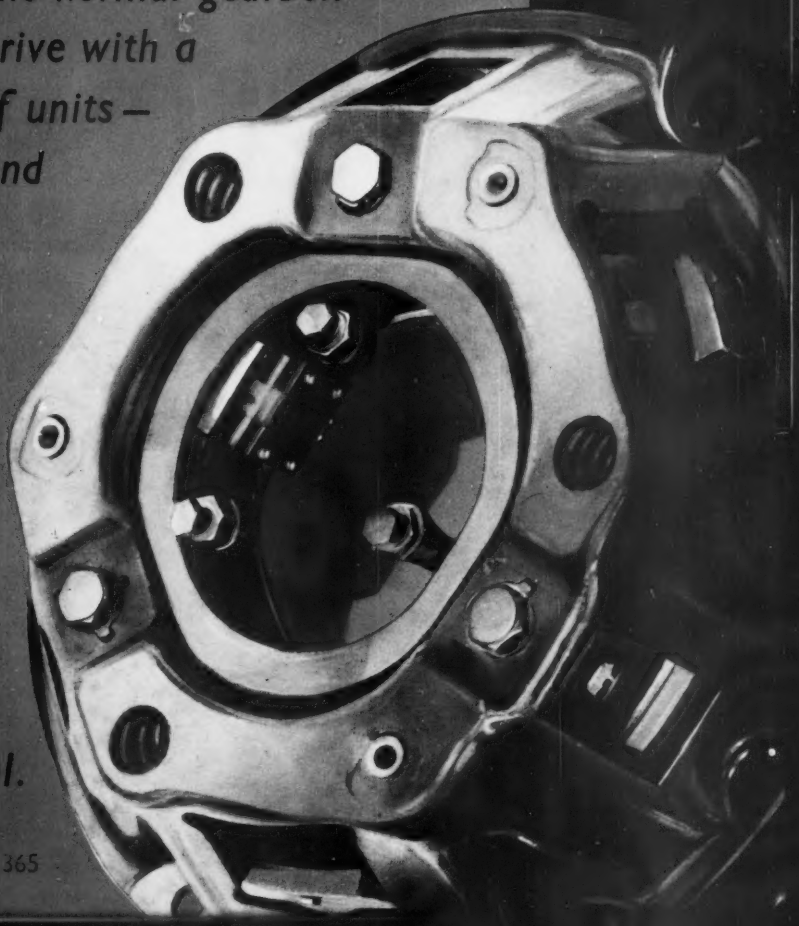

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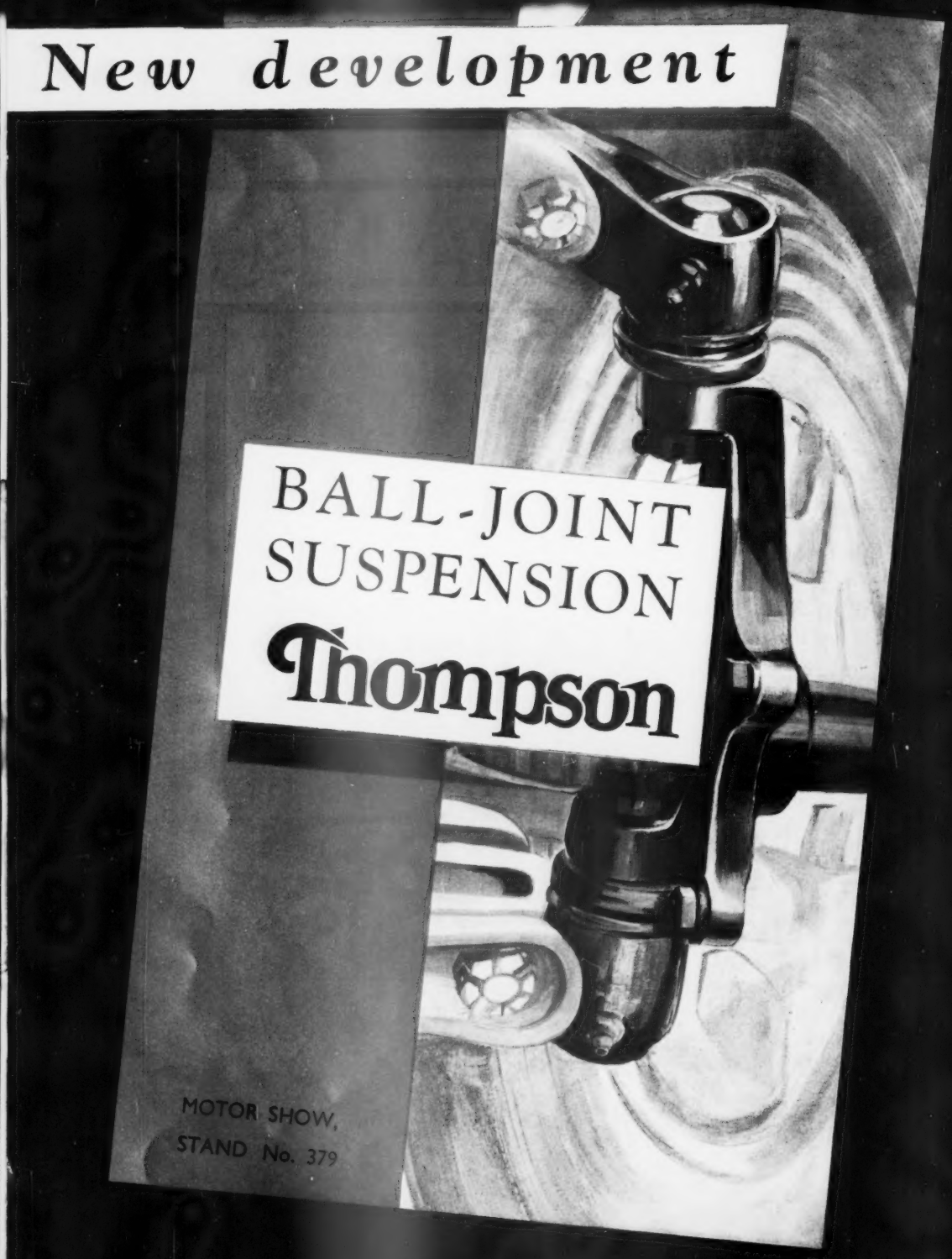

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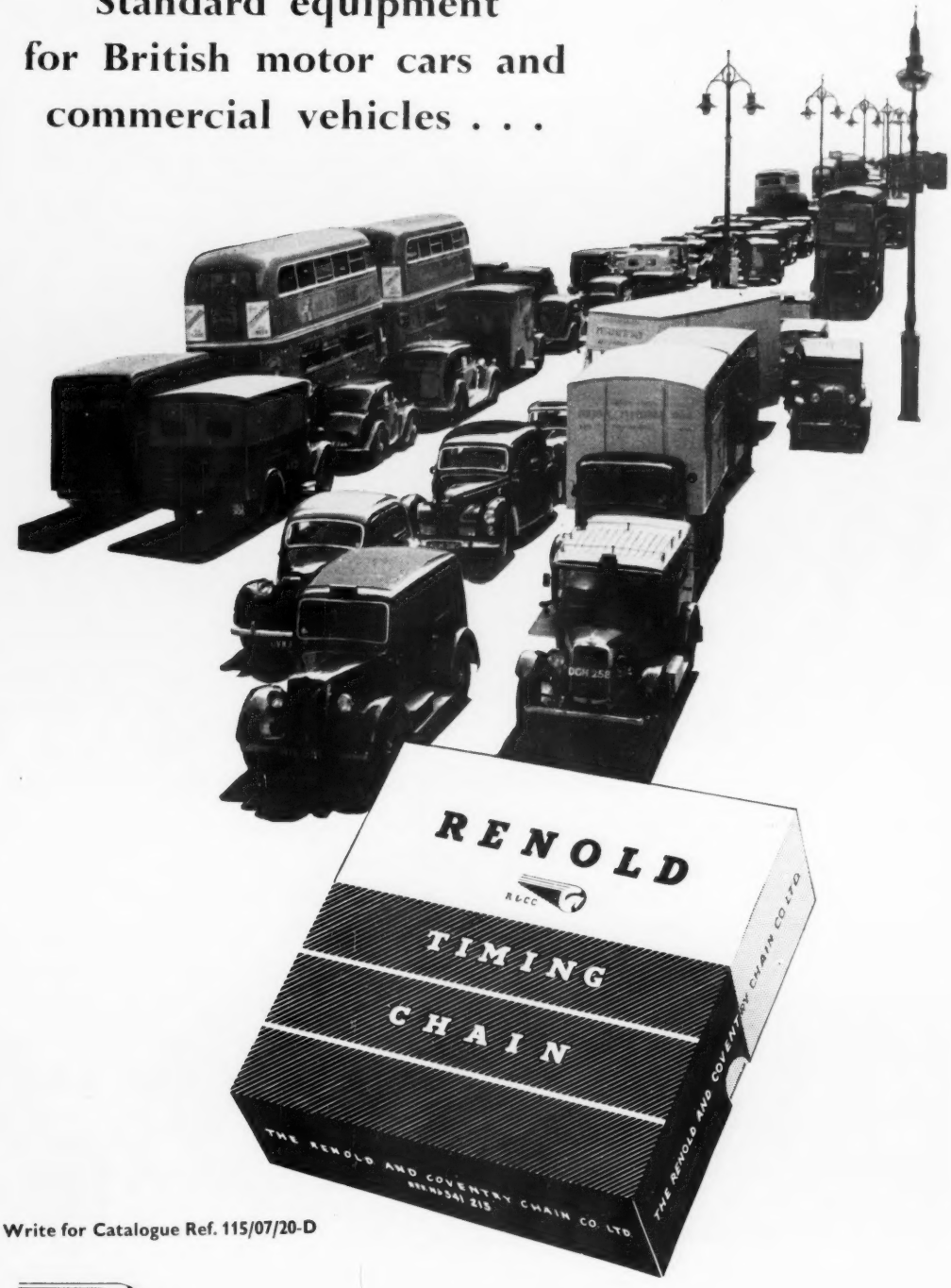

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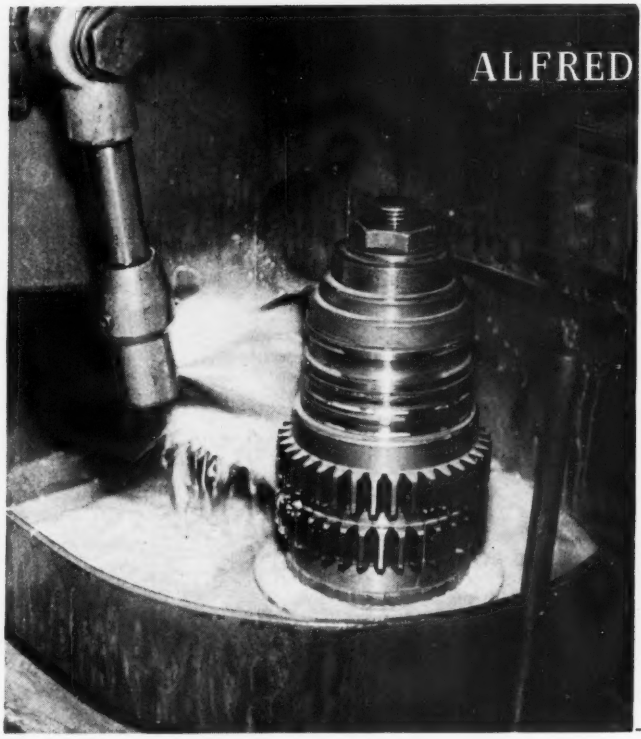

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the new water-miscible oil designed to take the place of neat metal cutting oils.

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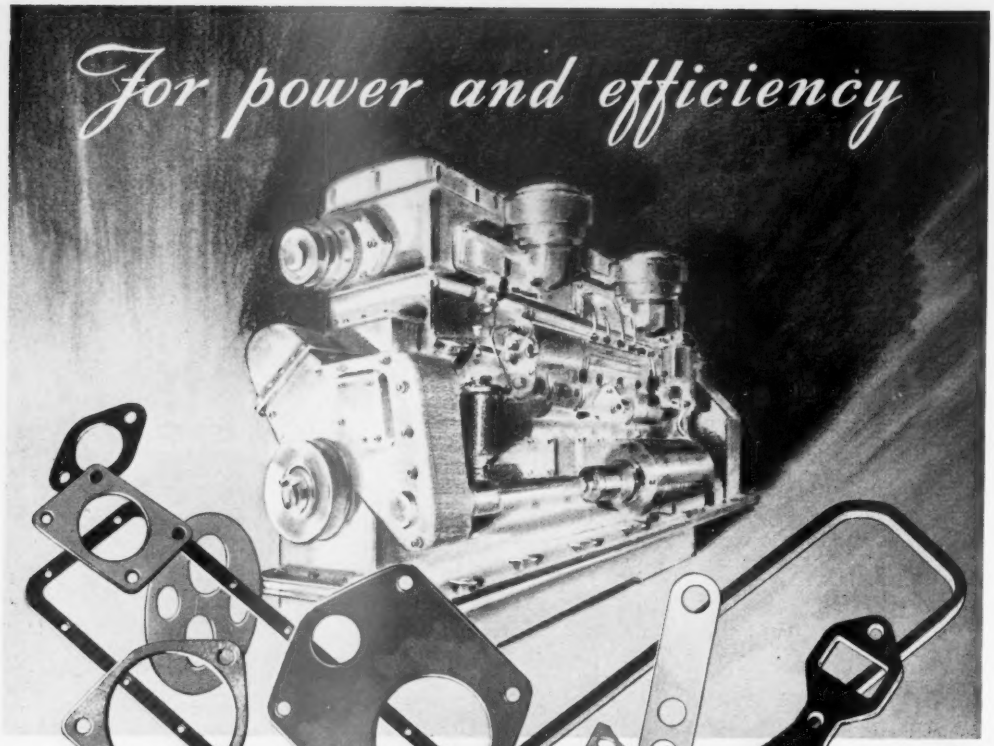

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Such is the report of Messrs. Leyland Motors Ltd. to whom the Bush was sent for examination. The report concludes: "The Tecalemit Syndromic Chassis Lubrication Equipment fitted to this machine has proved most effective."

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* *All other bearings lubricated by this system were found to be in the same perfect condition.*

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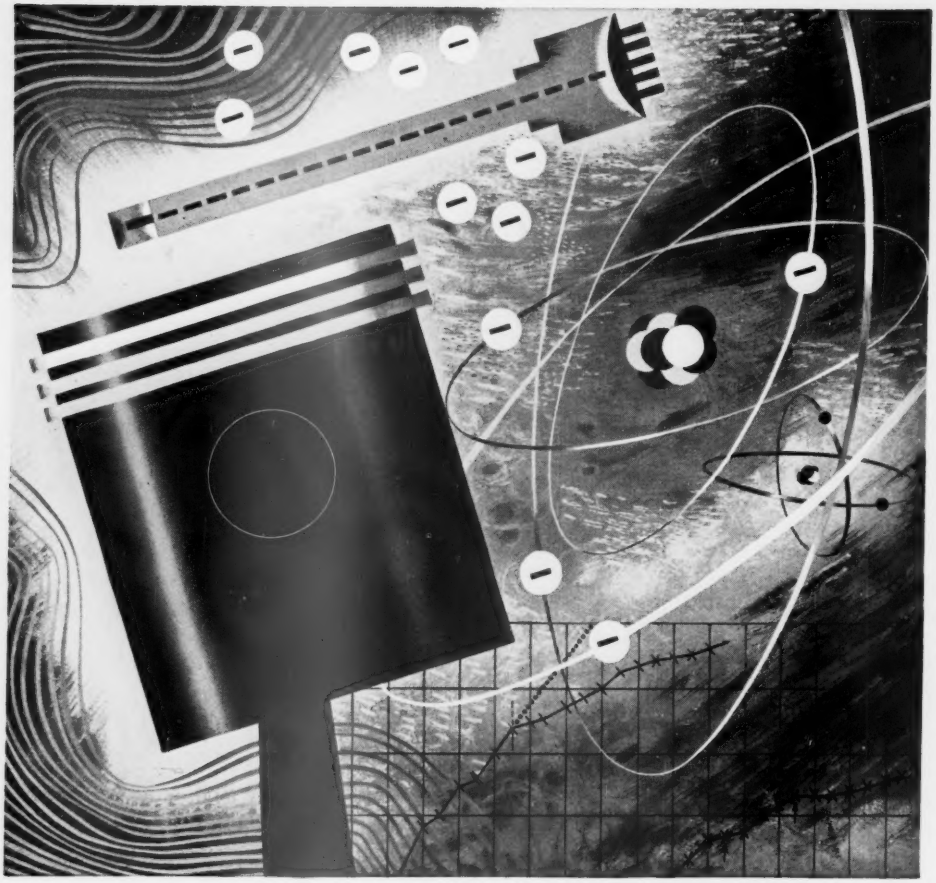

Atomic detectives

Metal wear is no new problem to engineers or research workers, but the work at Thornton has been transformed almost overnight by the application of radioactive isotopes to wear study. These are the atomic detectives of science They record the slightest finger-print of wear with inhuman speed and accuracy.

In a test now proceeding, a radioactive piston ring fitted in an experimental engine is supplying valuable data on the *rate* of wear. Wear on the piston ring removes minute particles of radioactive metal which are carried away by the lubricating oil. As the oil returns to the sump of the engine it passes a Geiger Counter, which automatically records the presence of radioactive metal. This fundamentally simple test is likely to give remarkable results, results that will be turned to practical account by Shell scientists and engineers.

In another test, a length of wire is drawn vertically through a tungsten carbide die which has been made radioactive. With the aid of radioactive tracers, which are transferred from the die to the wire, the radioactivity of the wire can be assessed by means of a Geiger Counter. By this method it is possible to measure 0.00000002 grams of die wear per foot length of wire.

These are but two examples of how work on the atom has given into the hands of Shell lubrication experts research aids that will give answers in weeks instead of years. To you that means even better performance from even better Shell lubricants. To industry as a whole it means a narrowing of the time-lag between the invention of new machines and the development of lubricants to suit them.

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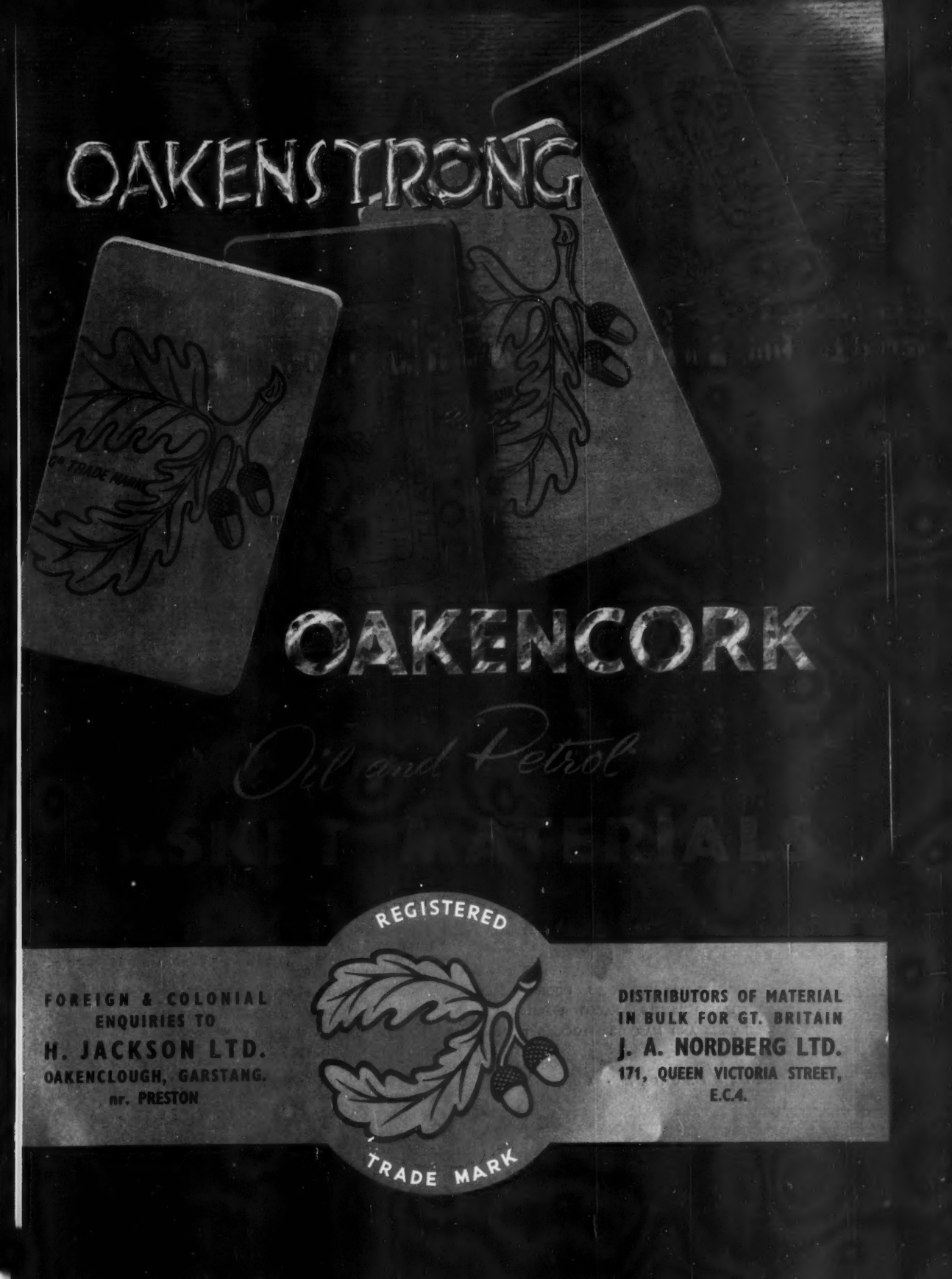

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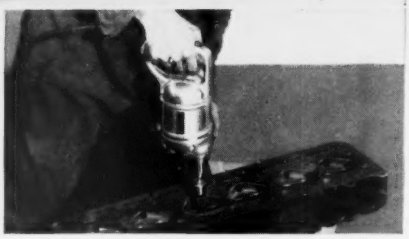
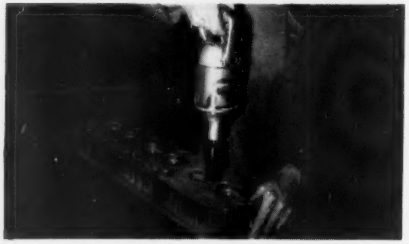


A ¼" B & D Drill and Carbon Removing Brush, rapidly and safely cleans cylinder head and valve ports.



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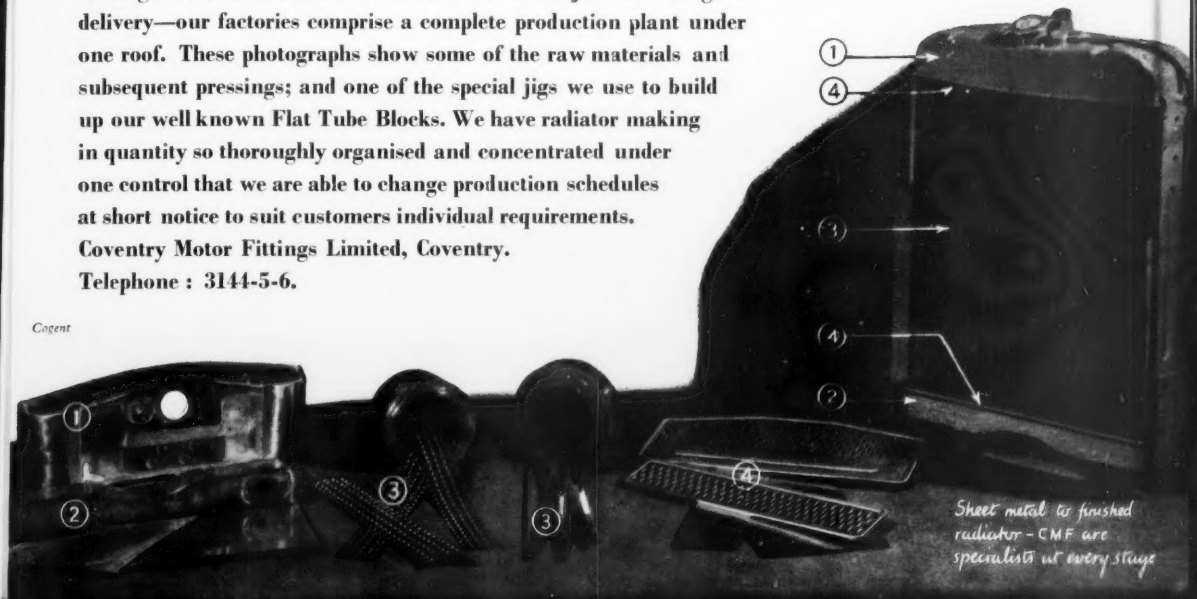

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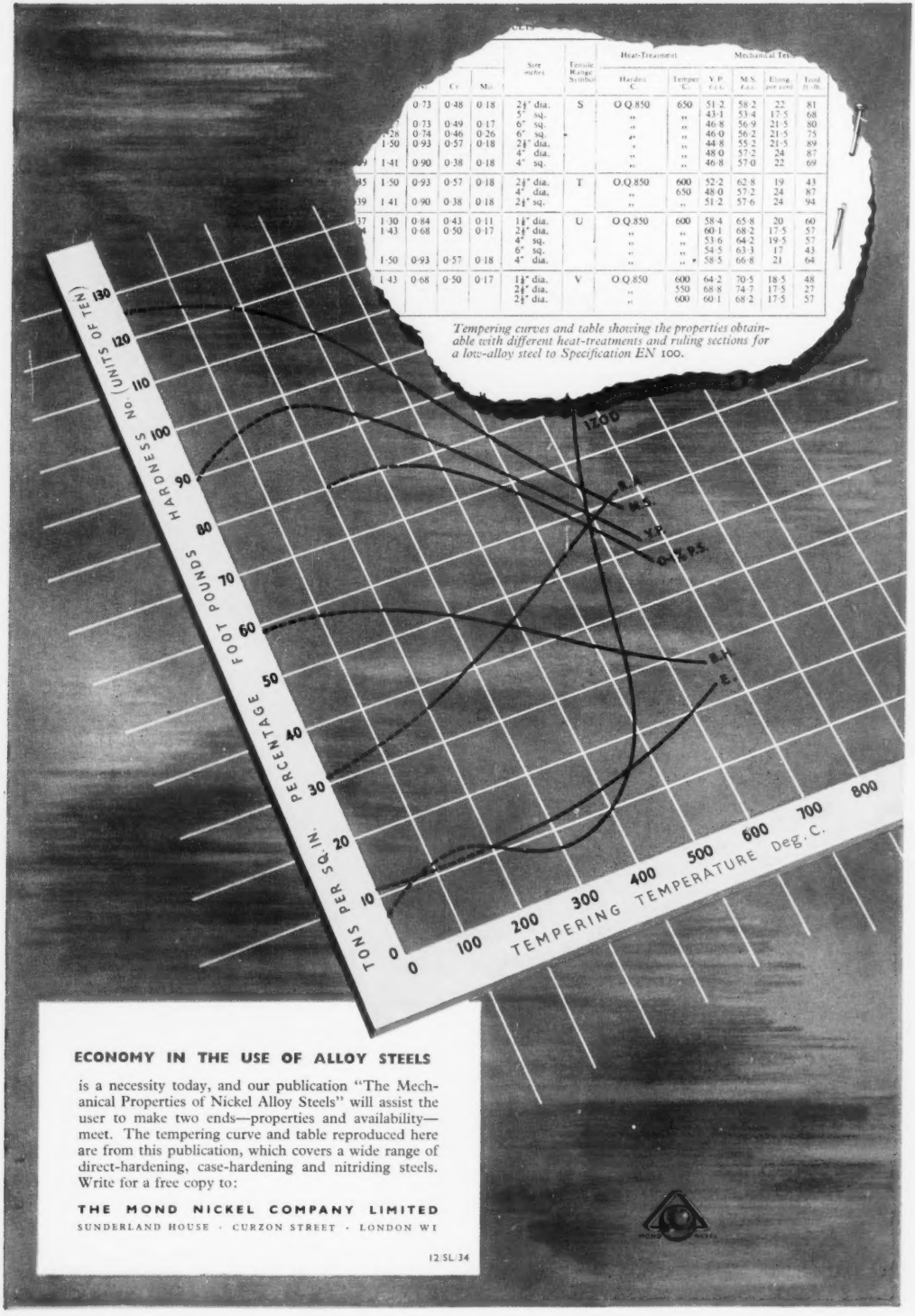

				Size inches	Tensile Range Symbol	Heat-Treatment		Mechanical Tests			
						Harden C.	Temper C.	Y.P. t.s.i.	M.S. t.s.i.	Elong. per cent	Izod ft.-lb.
		0·73	0·48	0·18	2½" dia.	S	O.Q.850	650	51·2	58·2	22	81
					5" sq.		,,	,,	43·1	53·4	17·5	68
		0·73	0·49	0·17	6" sq.		,,	,,	46·8	56·9	21·5	80
	0·28	0·74	0·46	0·26	6" sq.		,,	,,	46·0	56·2	21·5	75
	1·50	0·93	0·57	0·18	2½" dia.		,,	,,	44·8	55·2	21·5	89
					4" dia.		,,	,,	48·0	57·2	24	87
	1·41	0·90	0·38	0·18	4" sq.		,,	,,	46·8	57·0	22	69
	1·50	0·93	0·57	0·18	2½" dia.	T	O.Q.850	600	52·2	62·8	19	43
					4" dia.		,,	650	48·0	57·2	24	87
	1·41	0·90	0·38	0·18	2½" sq.		,,	,,	51·2	57·6	24	94
	1·30	0·84	0·43	0·11	1½" dia.	U	O.Q.850	600	58·4	65·8	20	60
	1·43	0·68	0·50	0·17	2½" dia.		,,	,,	60·1	68·2	17·5	57
					4" sq.		,,	,,	53·6	64·2	19·5	57
					6" sq.		,,	,,	54·5	63·3	17	43
	1·50	0·93	0·57	0·18	4" dia.		,,	,,	58·5	66·8	21	64
	1·43	0·68	0·50	0·17	1½" dia.	V	O.Q.850	600	64·2	70·5	18·5	48
					2½" dia.		,,	550	68·8	74·7	17·5	27
					2½" dia.		,,	600	60·1	68·2	17·5	57

Tempering curves and table showing the properties obtainable with different heat-treatments and ruling sections for a low-alloy steel to Specification EN 100.

ECONOMY IN THE USE OF ALLOY STEELS

is a necessity today, and our publication "The Mechanical Properties of Nickel Alloy Steels" will assist the user to make two ends—properties and availability—meet. The tempering curve and table reproduced here are from this publication, which covers a wide range of direct-hardening, case-hardening and nitriding steels. Write for a free copy to:

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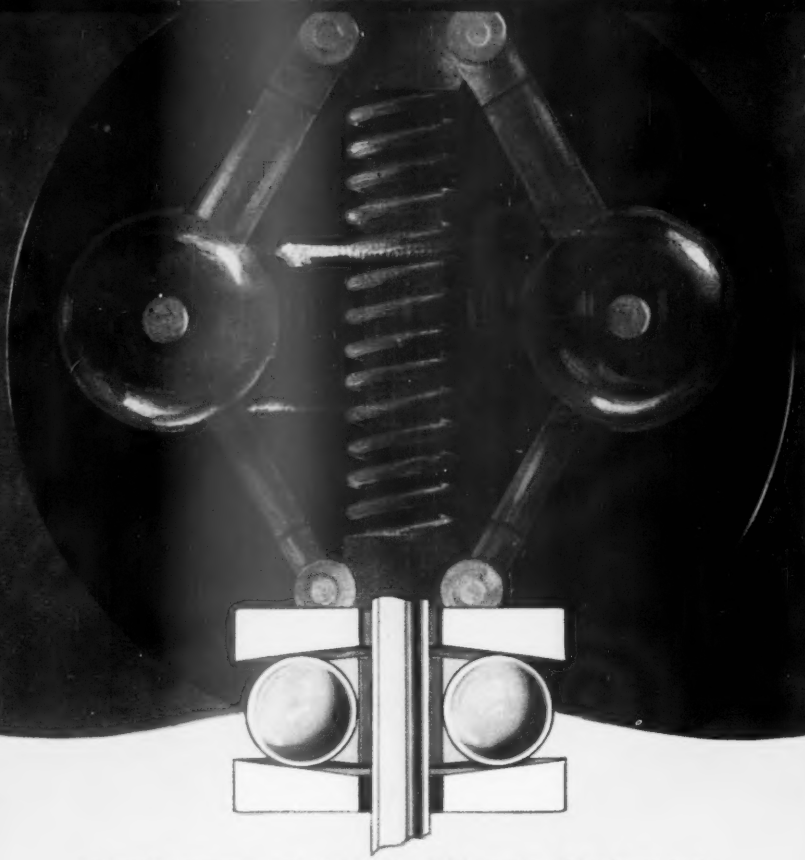

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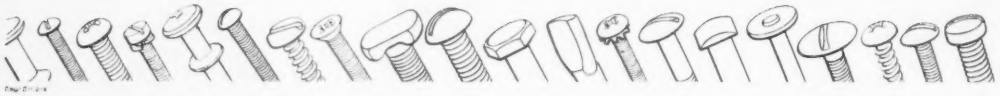

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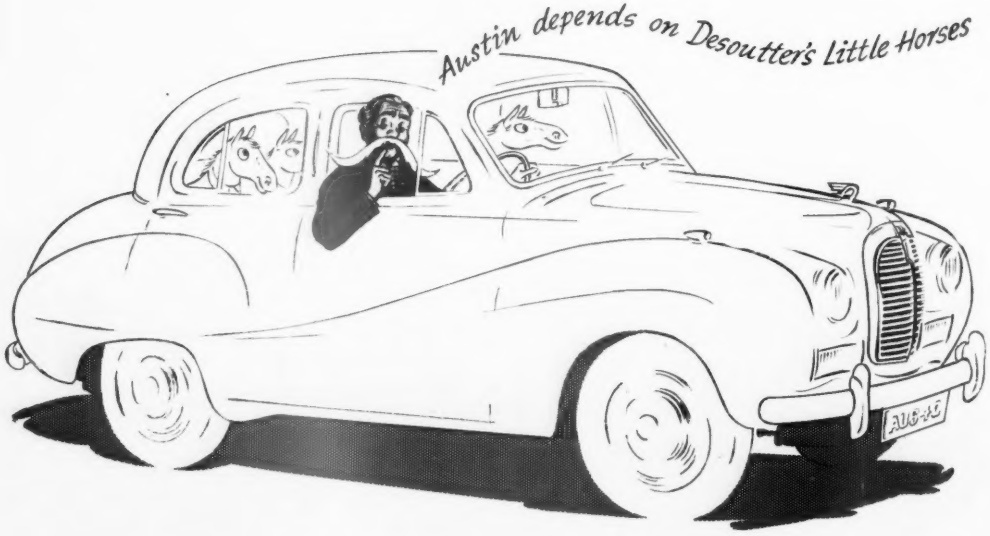

"The idea is" said Austin's Planning Department (very, very bright boys), "you put these bits and pieces on the moving conveyor and then along comes an engine and then a body and they all join up and you squirt oil and petrol in and drive off. See?" But what joins them up? What puts all the nuts and screws in the right place at the right time at the right pressure and before the darn thing moves on out of reach? Why, Desoutter Little Horses of course. Specially trained to work alone or in teams of 2, 3 or 4, guided and cared for by the nicest lot of engineers you could hope to meet, the little chaps do the whole job with certain and absolute precision.
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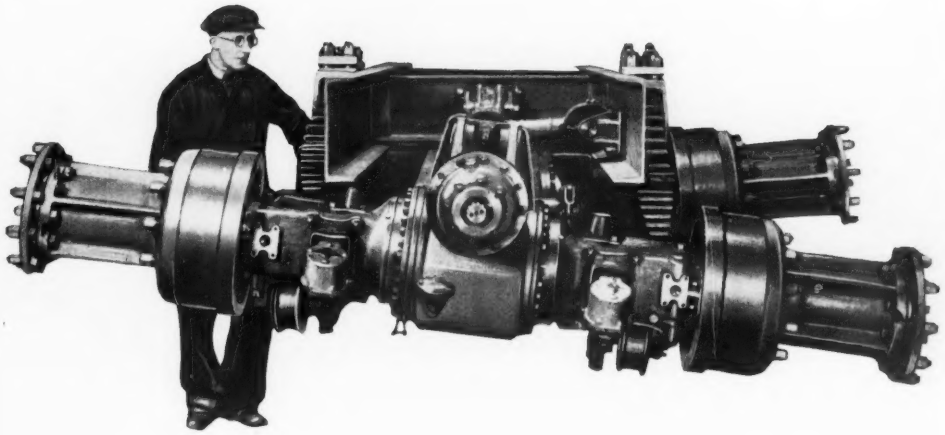

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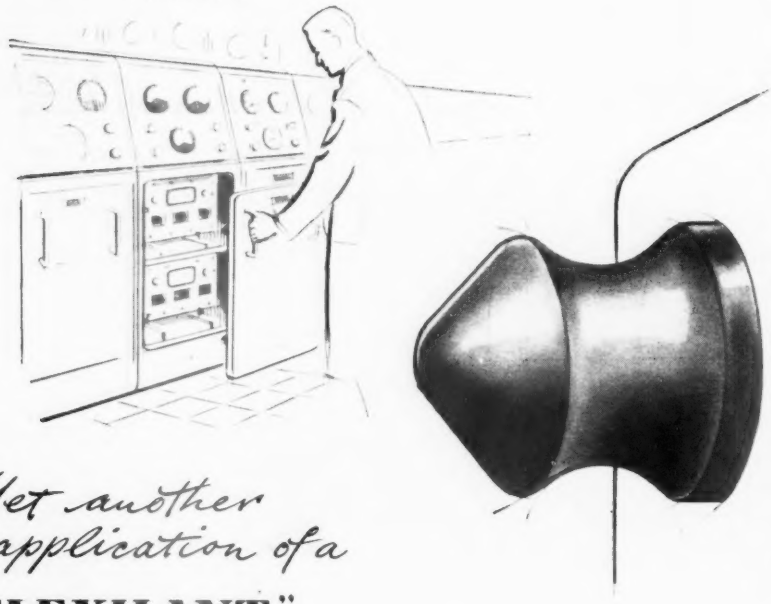

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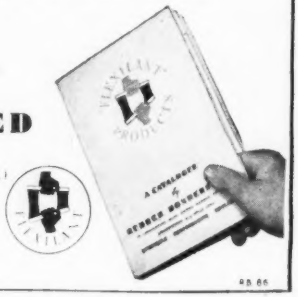

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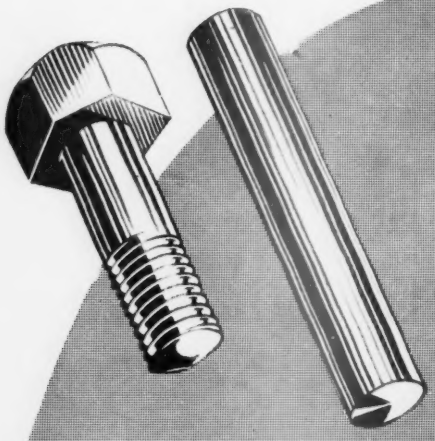
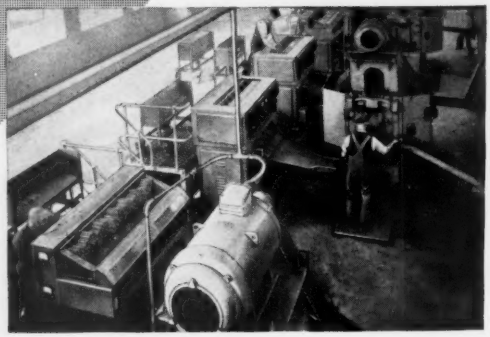


product	$\frac{3}{4}$" black bolts.
process	heating one end of pin for heading.
output	1,400 pins per hour.
work cost	8s. 6d. per 1,000 pieces.
equipment	Birlec 80 kW automatic induction heater.



There are now 10 Birlec units in this black bolt shop.

Three station, hand-fed, unit for heating bars.

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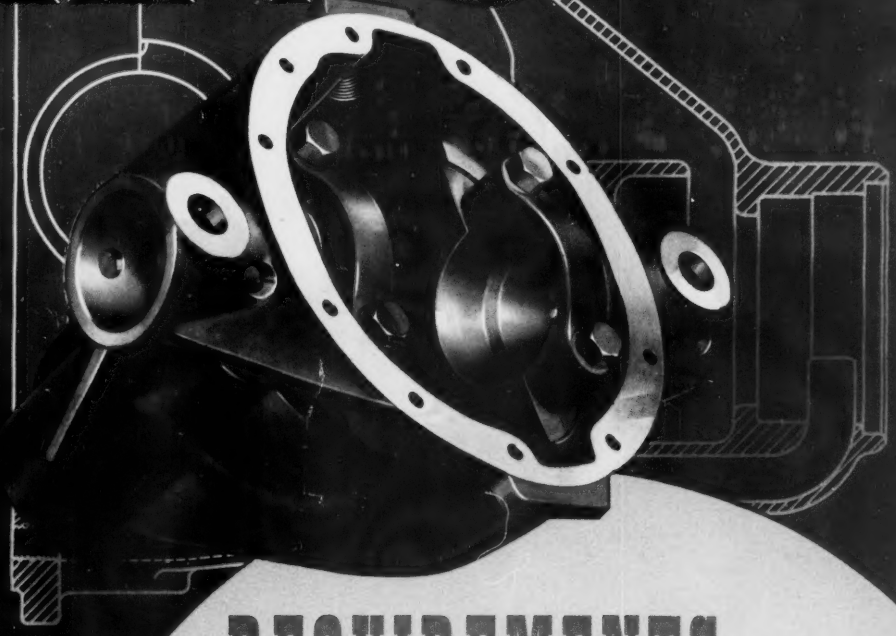

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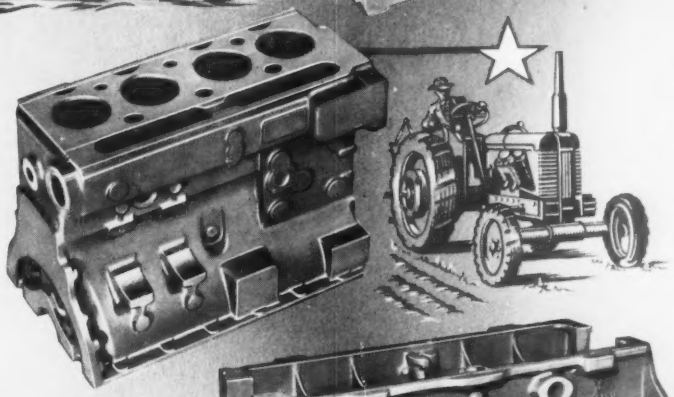

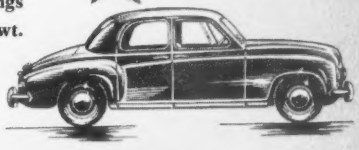

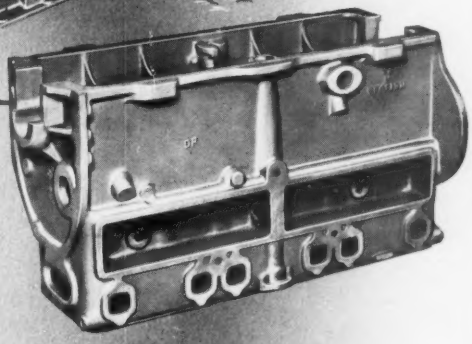

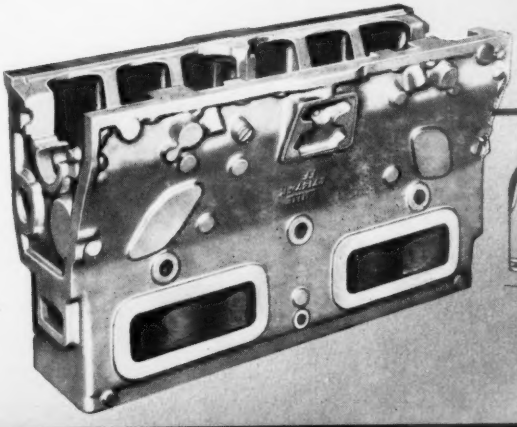

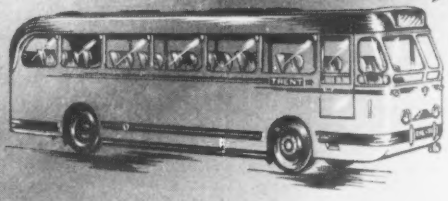

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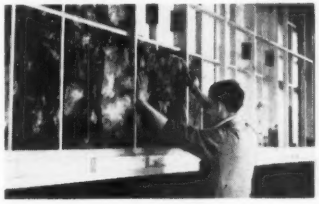

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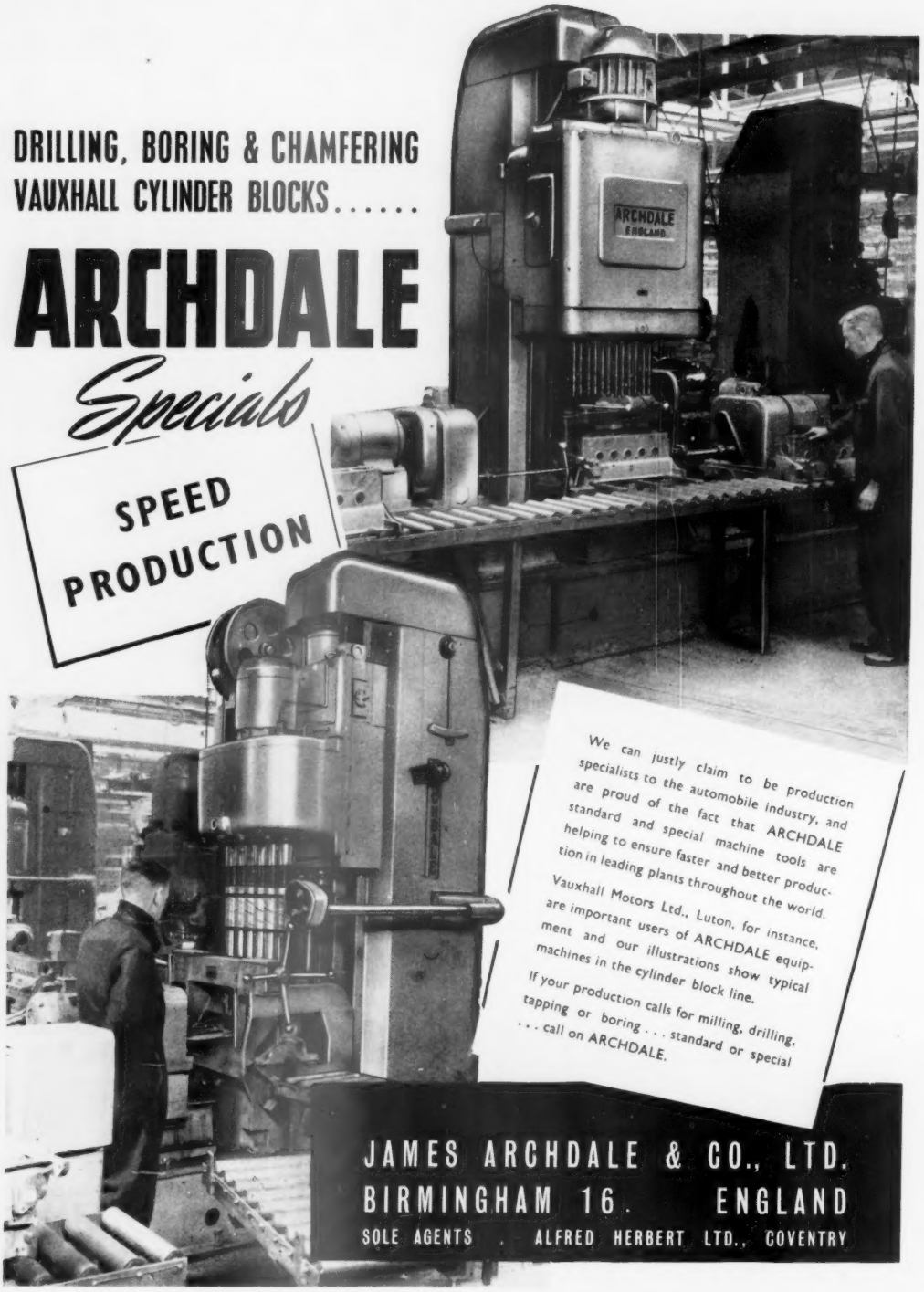

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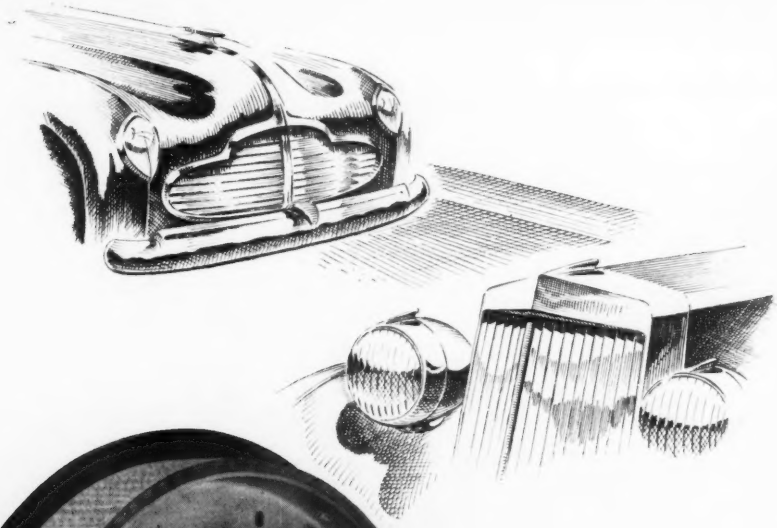

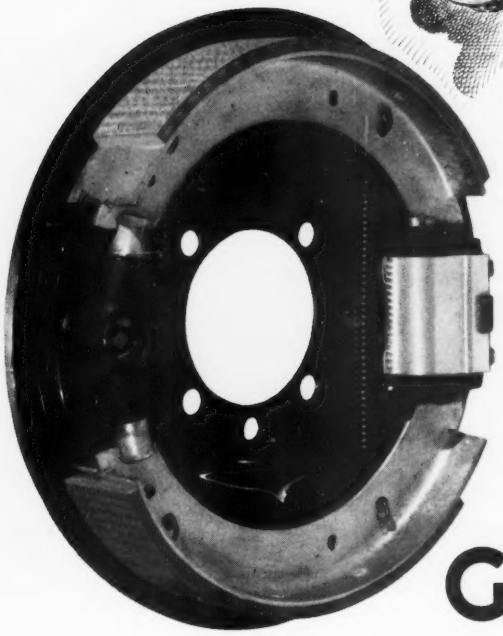

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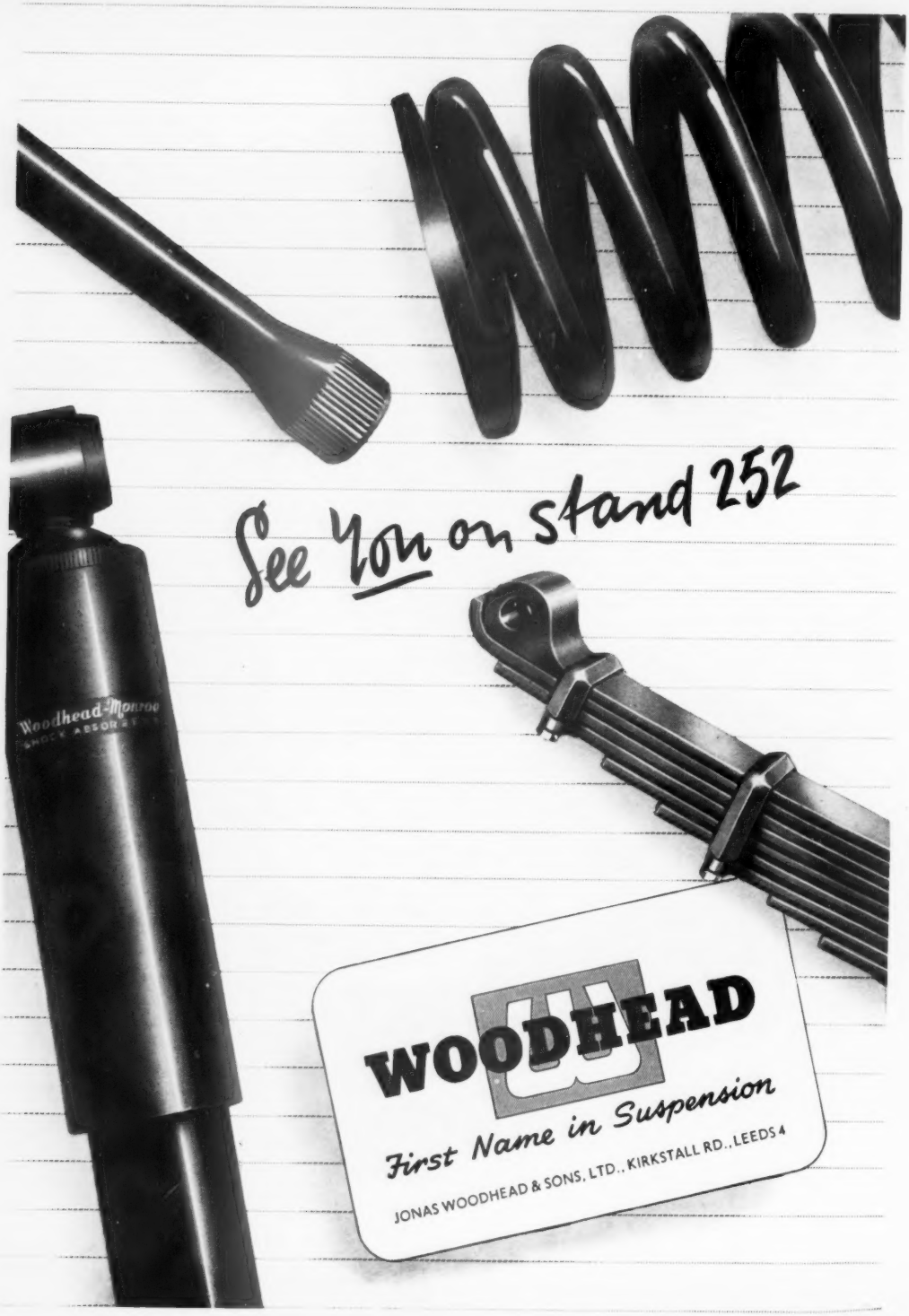
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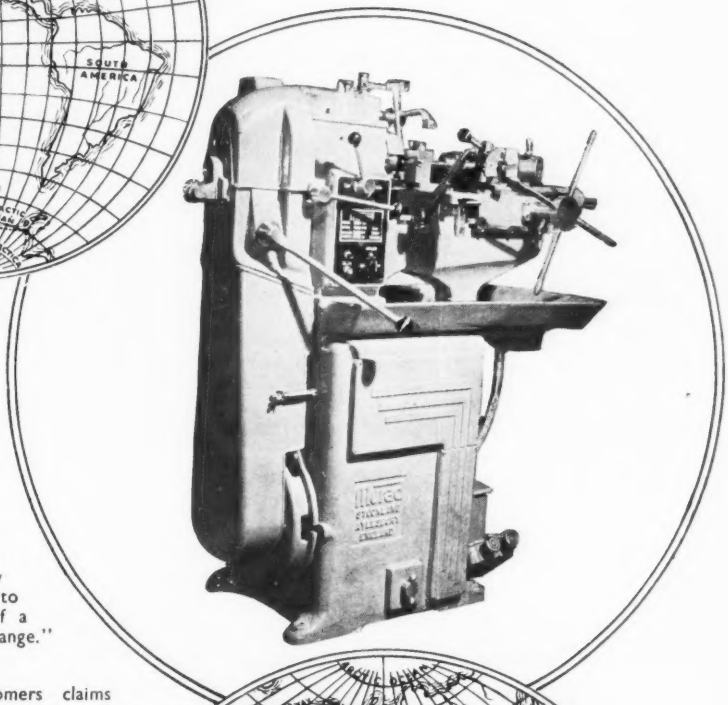

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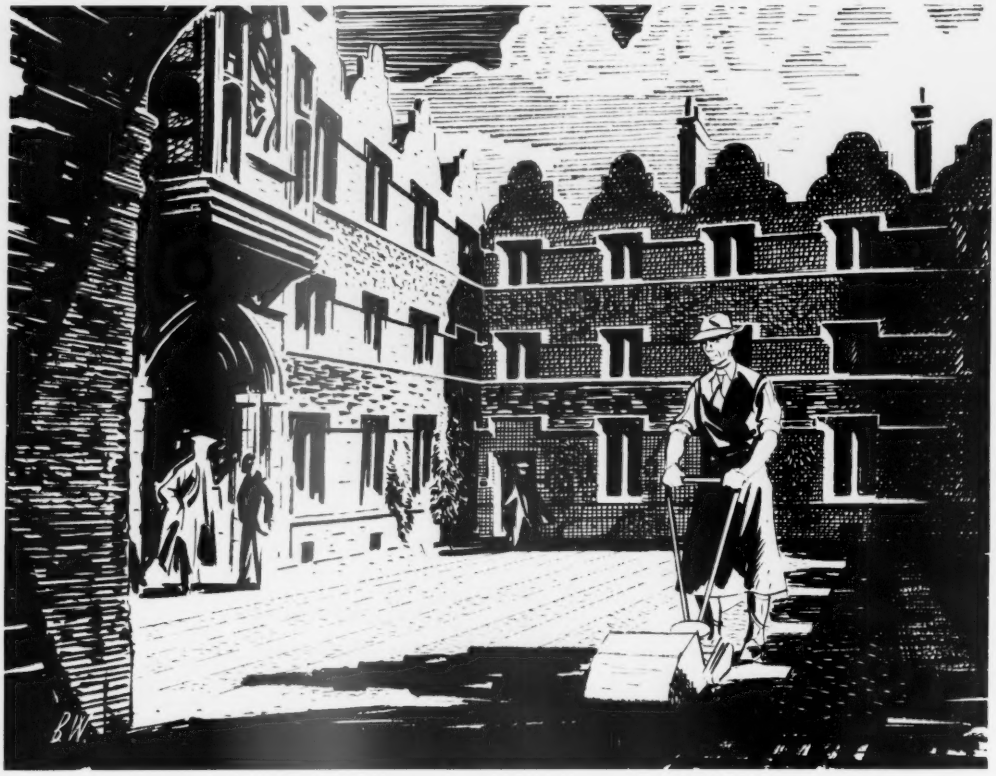

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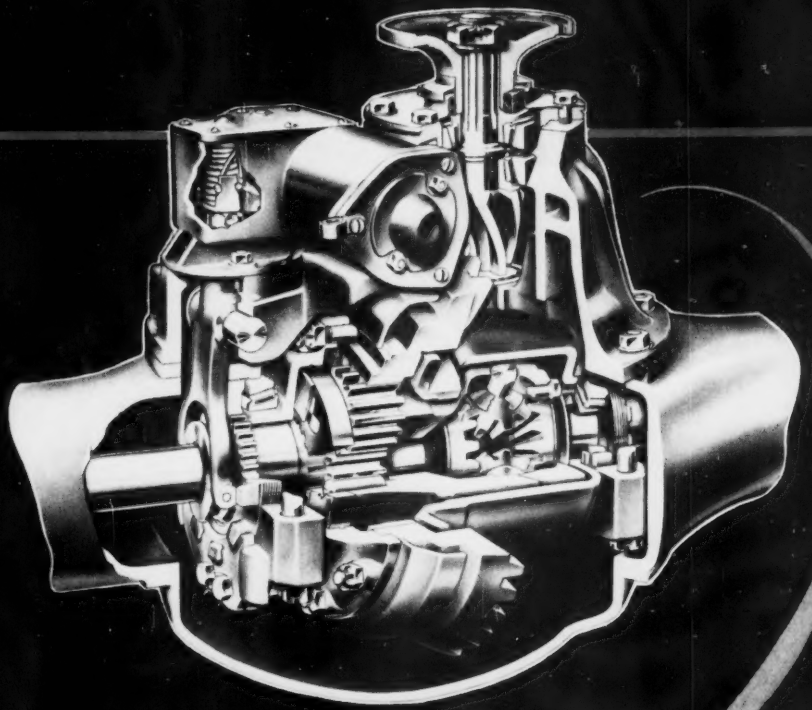

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THE name CAPASCO has been permanently associated with the development and perfection of Heavy Duty Moulded Brake Linings right from the start. CAPASCO pioneered the way, and is still leading this highly specialised field. CAPASCO Moulded Brake Linings are daily proving their efficiency and reliability under the most arduous conditions and in all forms of Transport.

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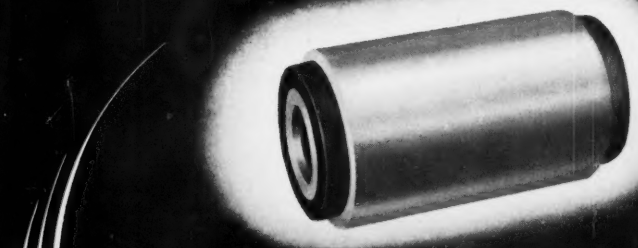
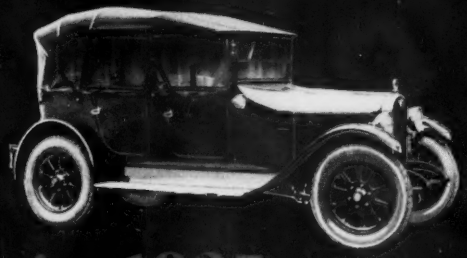


SILENTBLOC



The 1925 Austin 12 in which Silentbloc Bearings were first fitted in 1927

First in 1927 with the unlubricated bearing, Silentbloc maintain the lead in the design and manufacture of Flexible Components for the Motor Industry

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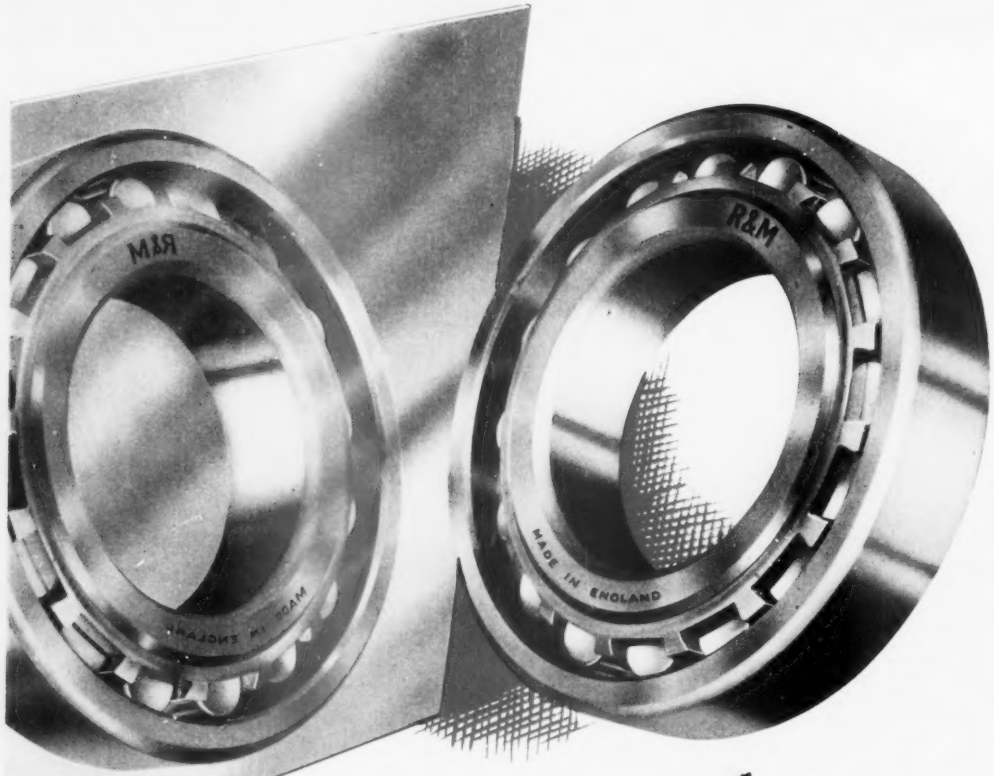

A finish to reflect upon

When you think of it, in a roller bearing unless the contact surfaces have supreme accuracy and mirror finish, then the life of the bearing must be curtailed.

In the manufacture of R&M Ball and Roller Bearings all the running surfaces are ground with unhurried care on the most modern types of precision grinders to superfine accuracy and finish.

This is just one of the many phases of skilful manufacture which has earned for R&M Bearings a reputation for frictionless efficiency, smooth running and long service.

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RANSOME & MARLES
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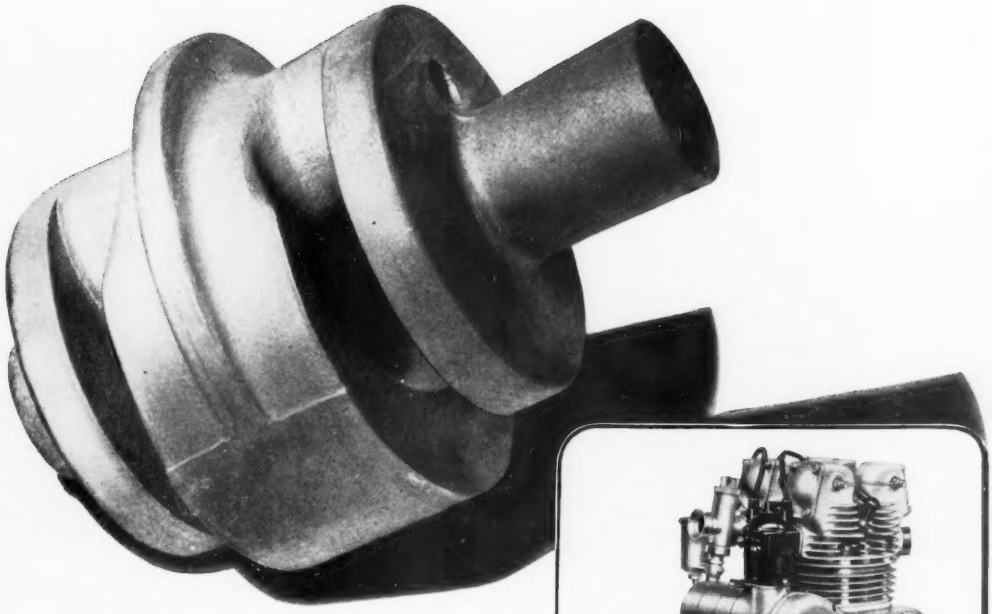

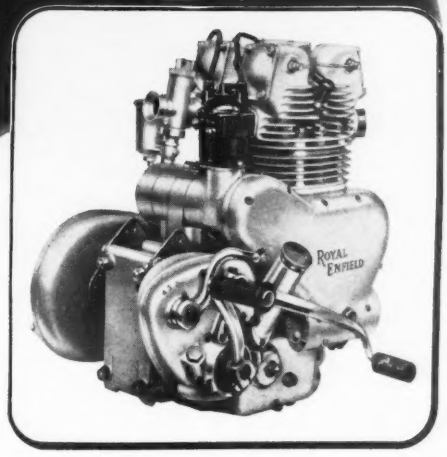

The engine of the Royal Enfield "500 Twin" motor-cycle in standard trim develops 25 B.H.P. at 5,750 R.P.M. and can reach 7,000 R.P.M. The crankshaft is a "Harper-Meehanite" casting. The Enfield Cycle Co. Ltd. say: "The excellent machining properties of the material, together with its mono-bloc construction, make this a much simpler production proposition than a built-up crankshaft, besides ensuring that it runs dead true at all times."

Harper quality covers grey iron and Meehanite castings, and also metal pressings, machining, enamelling and other finishes and sub-assembly work.

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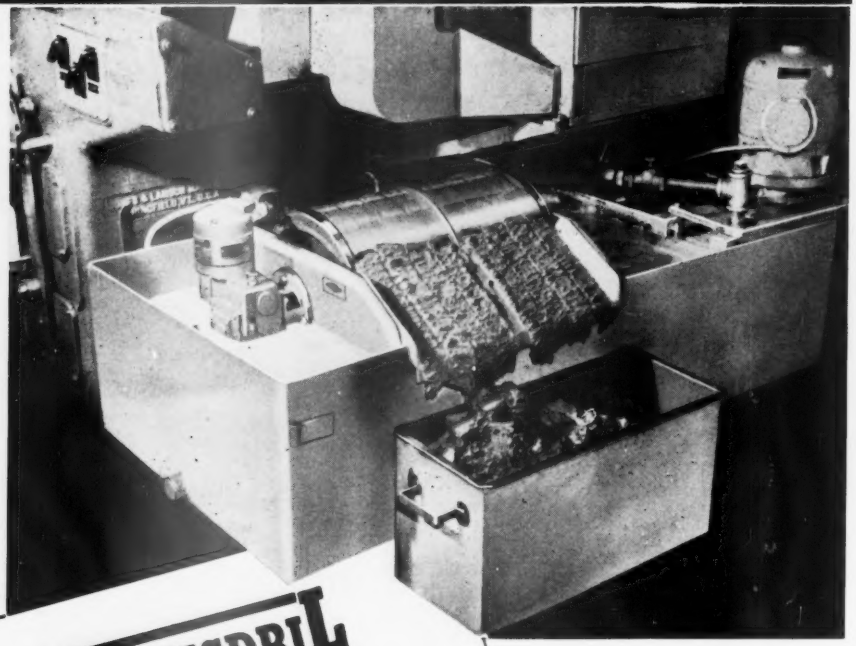

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Imp. Galls. Per min.	8	16	33	58	83

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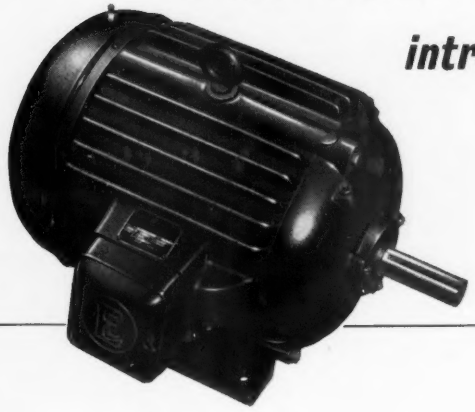
introducing standardisation



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These motors comply fully with B.S.168 and draft specification CN (ELE) 6814 of the British Standards Institution.

Dimensions are similar to those laid down in the American NEMA Standard MG1-1949.

For a similar frame size the horse-power outputs are increased to accord with the generally accepted British practice.

Frame B.225 is illustrated.

'ENGLISH ELECTRIC'

industrial motors

THE ENGLISH ELECTRIC COMPANY LIMITED, QUEENS HOUSE, KINGSWAY, LONDON, W.C.2

Industrial Motor Works, Bradford

WORKS: STAFFORD · PRESTON · RUGBY · BRADFORD · LIVERPOOL · ACCRINGTON

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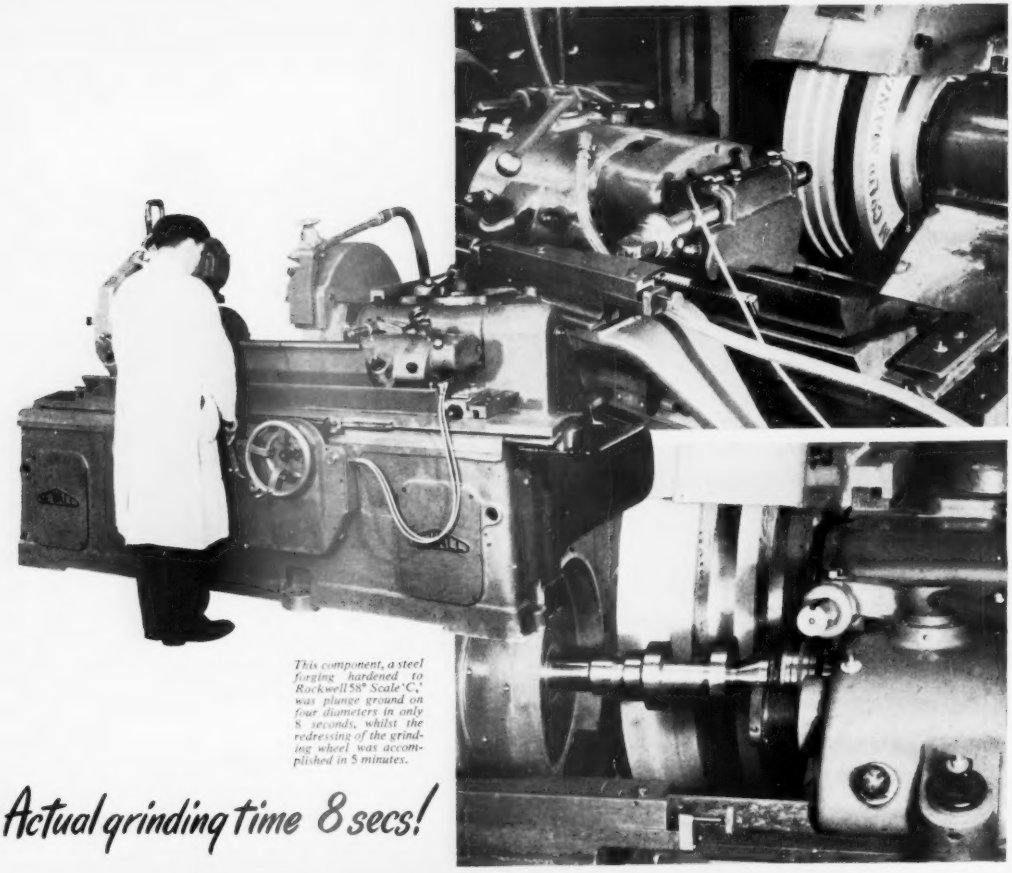
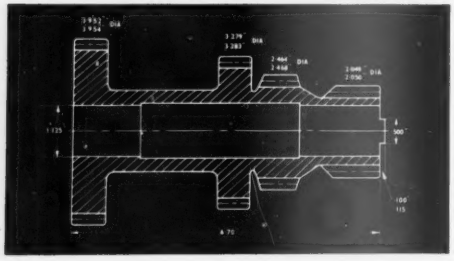


This component, a steel forging hardened to Rockwell 58° Scale 'C,' was plunge ground on four diameters in only 8 seconds, whilst the redressing of the grinding wheel was accomplished in 5 minutes.

☆ *Actual grinding time 8 secs!*

Grind all diameters at one plunge..

In common with other leading automobile manufacturers, the Ford Motor Co., (Canada) Limited employ only the most modern production methods. Amongst their tools is this Newall 10″×24″ L.A. Grinding Machine. With this machine, all diameters of Gear Transmission Countershafts can be ground at one plunge—the actual grinding time taken being only 8 seconds. The L.A. Grinding Machine has proved to be the right tool for high and economic outputs.



◀ *gear transmission countershaft*

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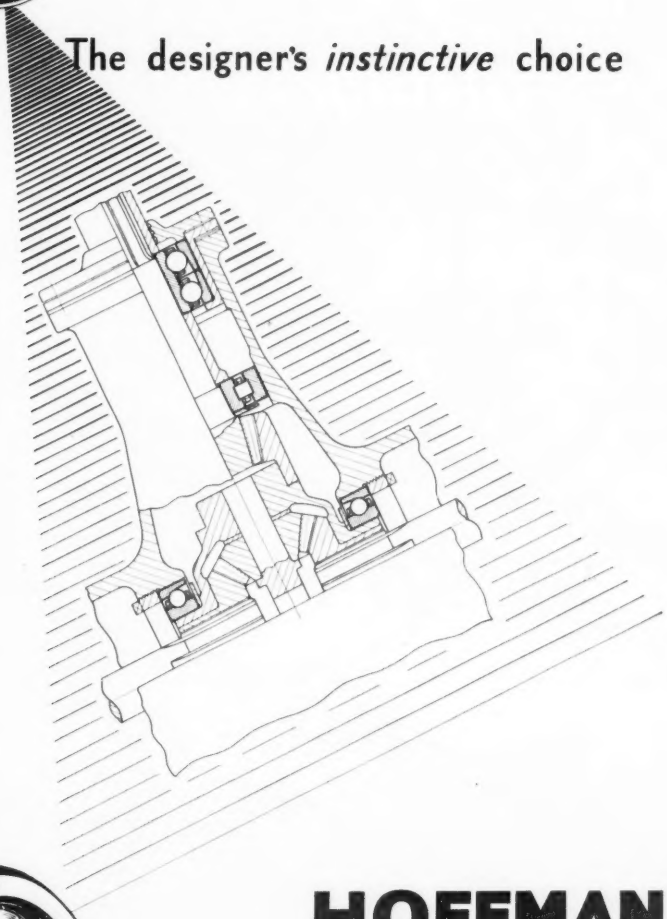

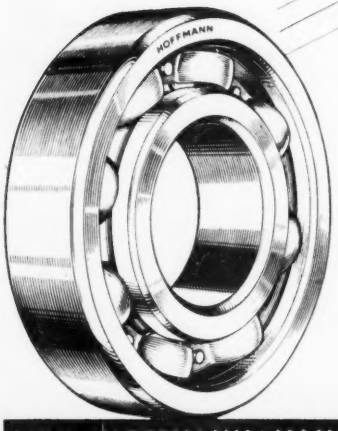

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Robustness and reliability are essential for the trouble free performance of the back axle bearings. The wearlessness of the Hoffmann Bearings ensures constant accurate gear mesh resulting in long life and quieter running.

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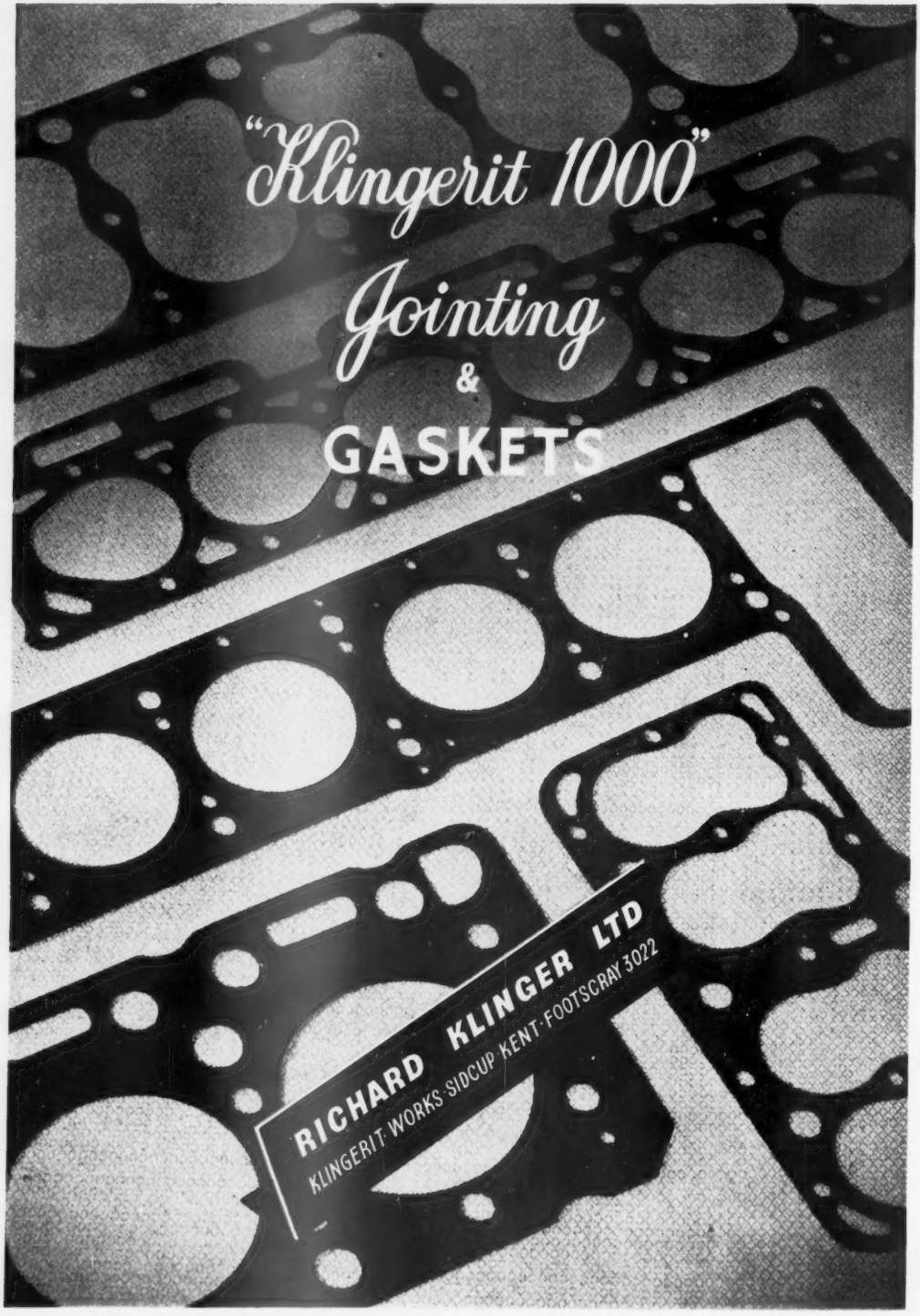
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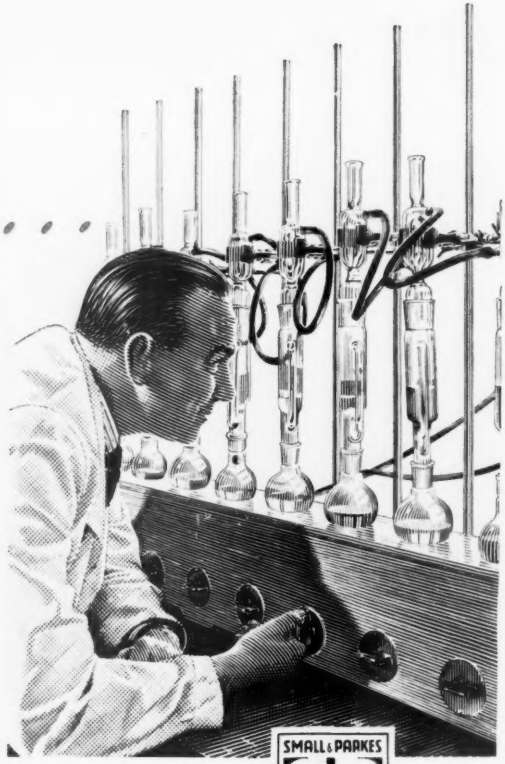

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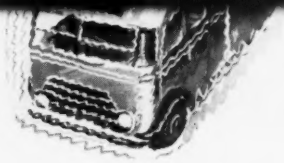

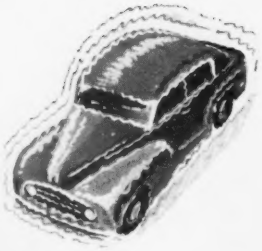

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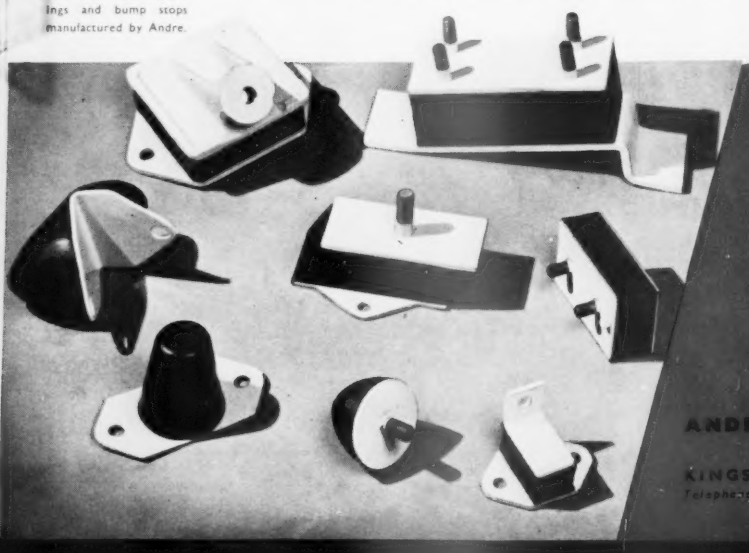

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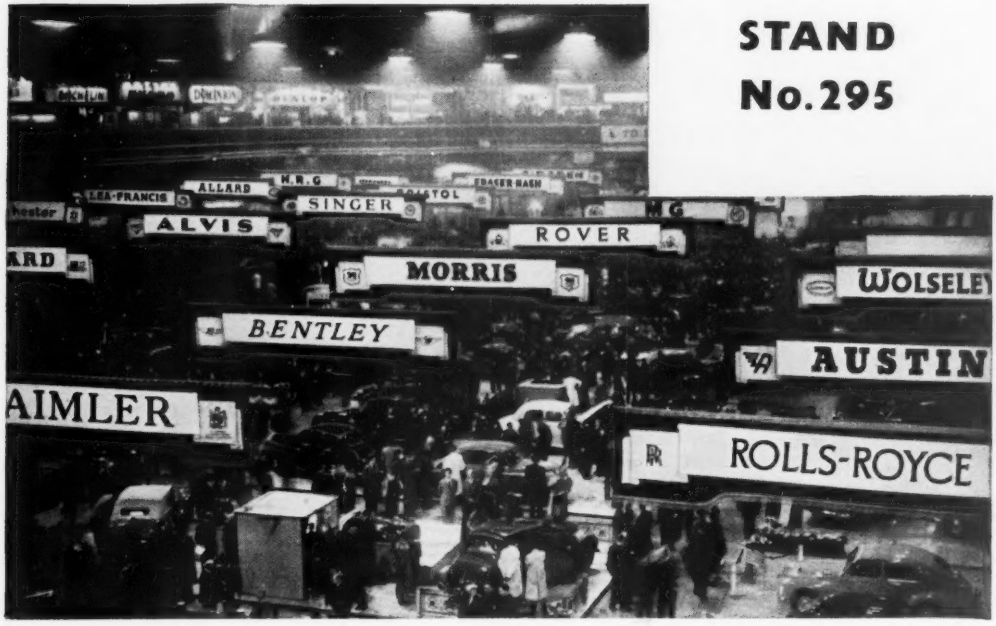

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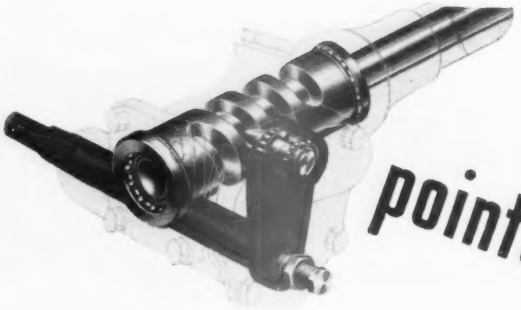

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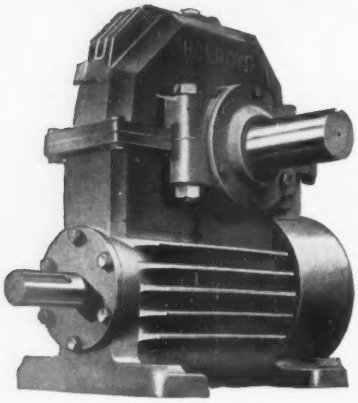
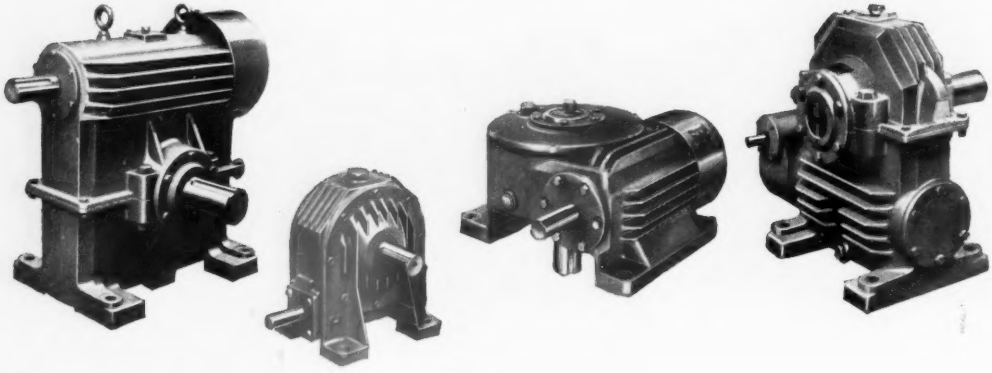


meet the boys . . . Some of our customers tell us they don't know how many types of Totally Enclosed Worm Reduction Units we make. Well! Here they all are. The big fellow at the top is our Underdriven (Standard) Worm Reduction Unit—comes in any size from 3″ to 24″ centres and ratios from 5 1 to 70 1. Then, first on the left below, our Overdriven W.R.U. — same range of centres and ratios but with driving shaft at top. Little Tich is our F. Type for fractional horsepower — 3 sizes: $1\frac{1}{2}$″, $1\frac{3}{4}$″, and $2\frac{1}{2}$″ centres each with 3 different relative positions of the input and output shafts. It may be mounted with the feet in any plane. The fat flat chap is our Vertical Type—same range as the Standard, but with a vertical wheel shaft that can be extended above or below the box. And last but far from least, our Double Reduction Unit for ratios up to 3000 1. Tough looking lot, aren't they! Tough, hard-working and reliable as a bevy of iron-jawed bank managers.

Holroyd Gear Reduction Units



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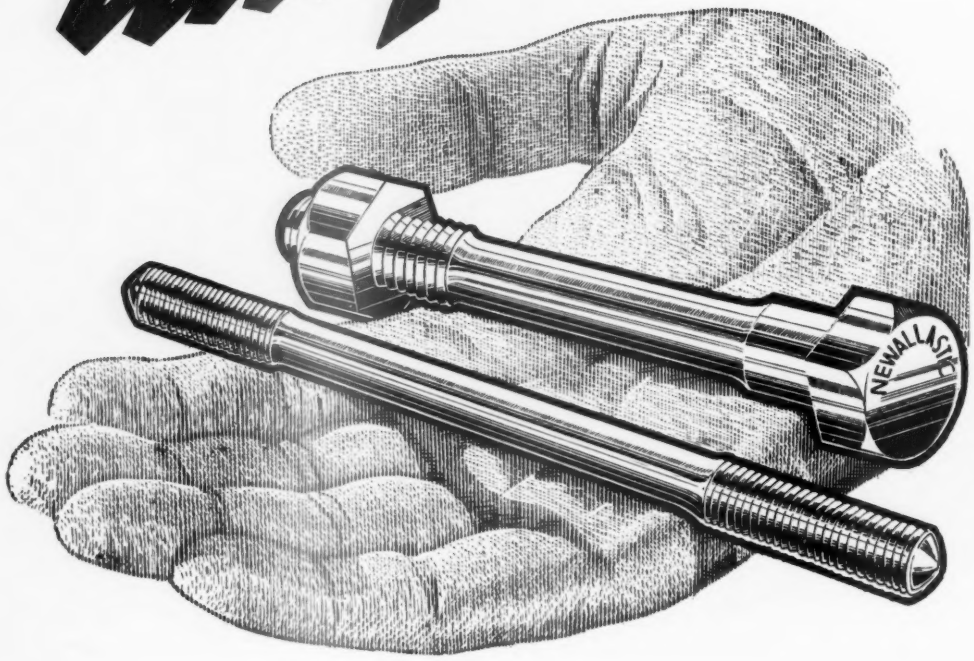
Unique



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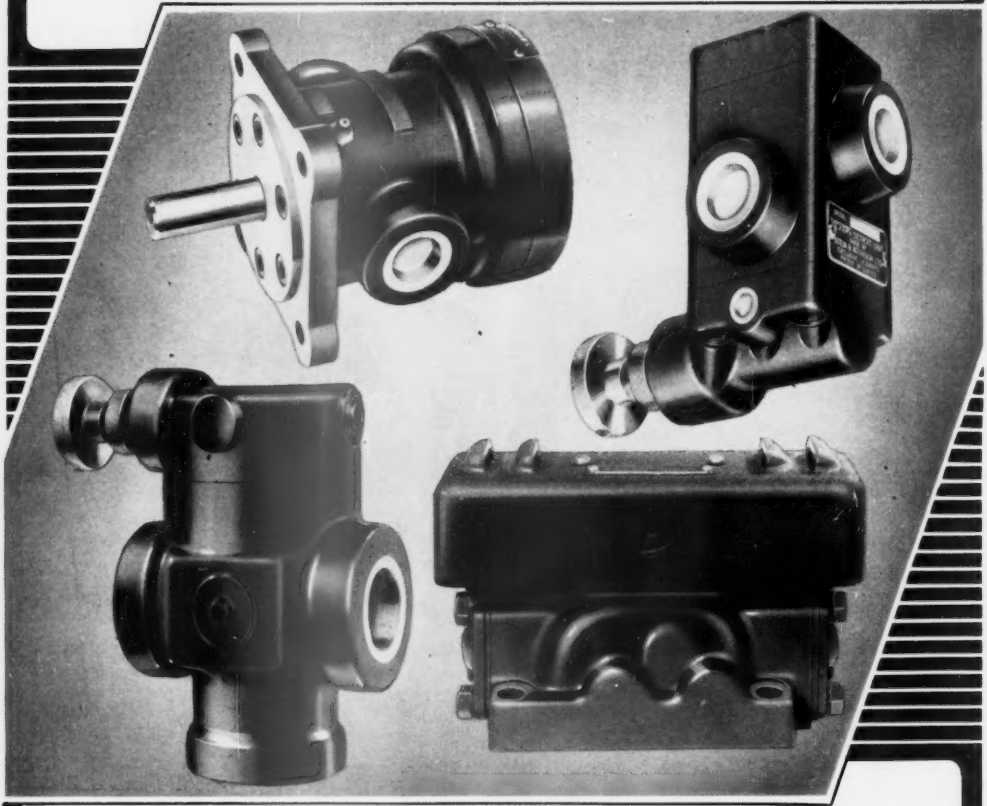

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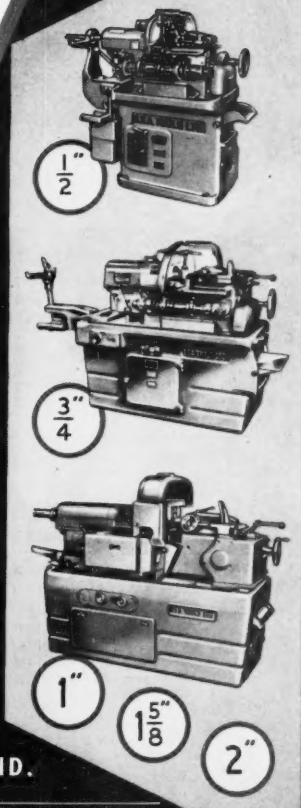

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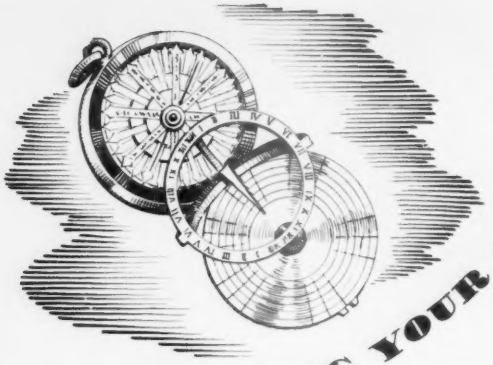

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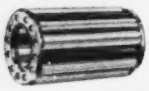
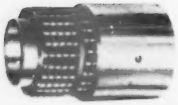
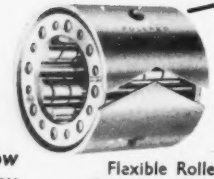


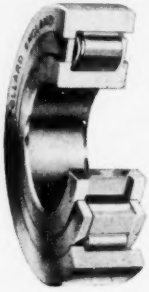


Flexible Roller Solid Race Bearing



Flexible Roller Split Race Bearing

"Max-Load" Solid Roller Bearing Separable Outer Race and Built up Cage



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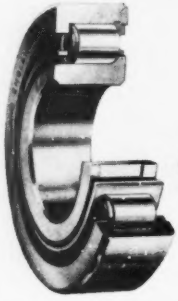
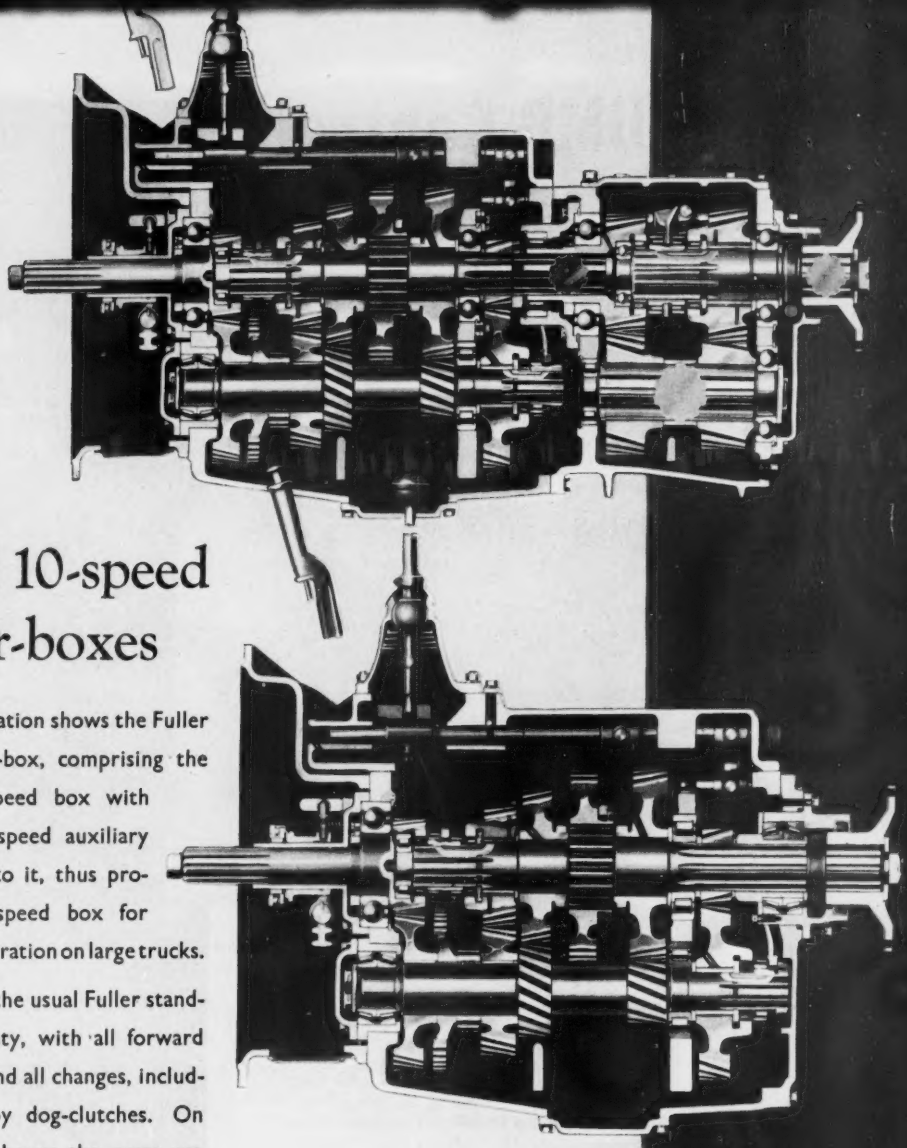


"Max-Load" Solid Roller Bearing Separable Inner Race and One Piece Cage



5 and 10-speed gear-boxes

The top illustration shows the Fuller 10-speed gear-box, comprising the famous five-speed box with a Fuller two-speed auxiliary box built on to it, thus providing a ten-speed box for heavy-duty operation on large trucks.

This unit is to the usual Fuller standard of high-duty, with all forward gears helical, and all changes, including reverse, by dog-clutches. On both of these boxes the gears are shot-peened and crown-shaved, to avoid stress concentration.

The lower illustration shows the Fuller 5-speed gear-box. Every gear is helical and engaged by dog-clutches and to reduce shaft deflection to a minimum the mainshaft is supported on three bearings, and the layshaft kept short, making one of the highest-duty gear-boxes ever produced.

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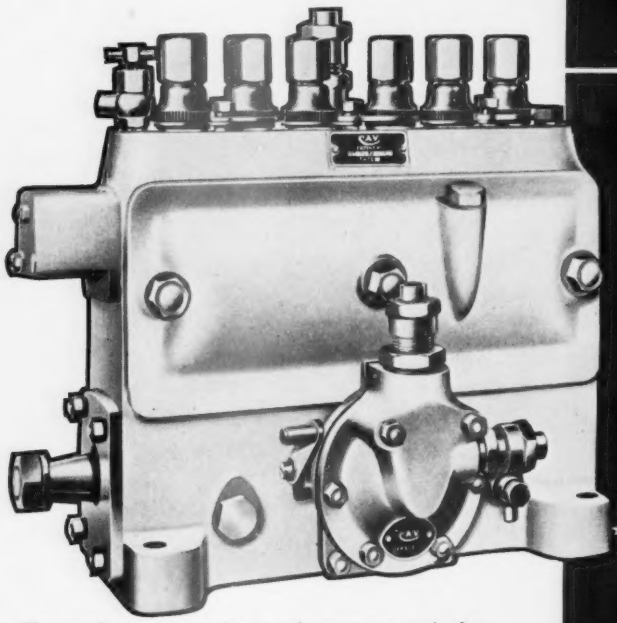

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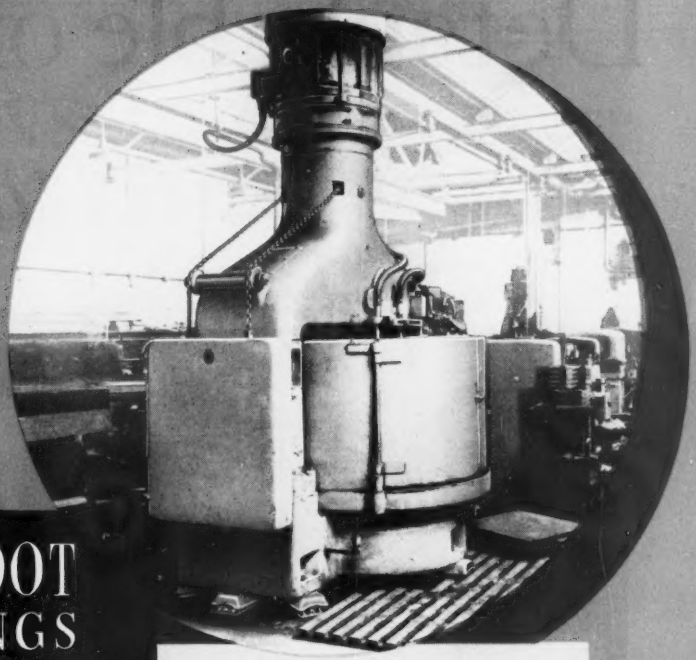

CUSHYFOOT MOUNTINGS

Increasing accuracy of grinder

This Lumsden Grinder, used by Mellor Bromley Ltd., Hosiery Machine Manufacturers, Leicester, is mounted on eight "Cushyfoot" units. In addition to preventing the transmission of vibration from one machine to another, the mounting has been found to eliminate wheel shudder.

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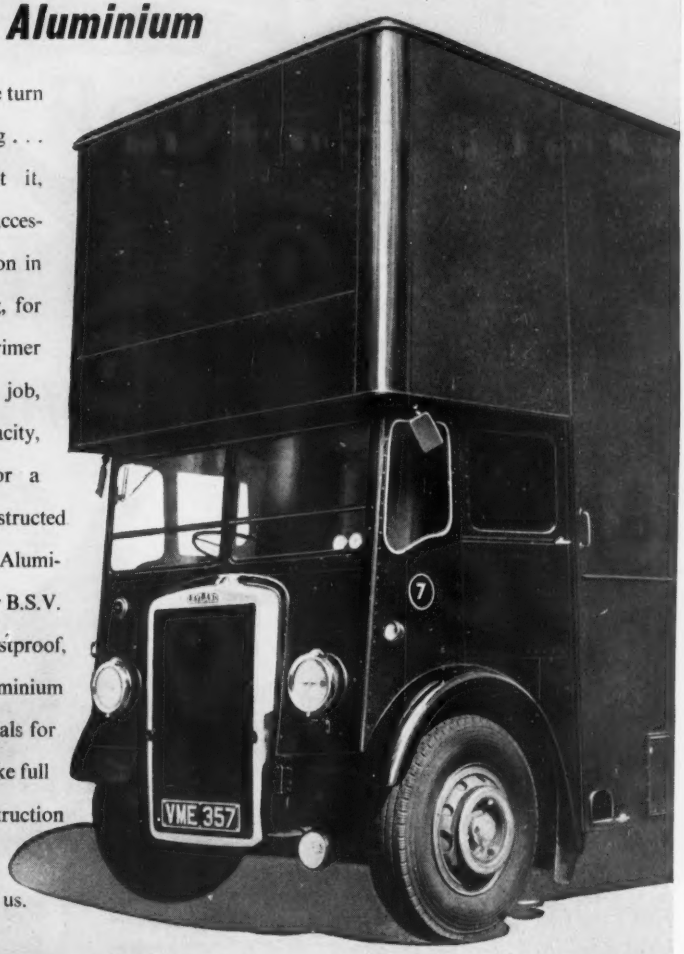

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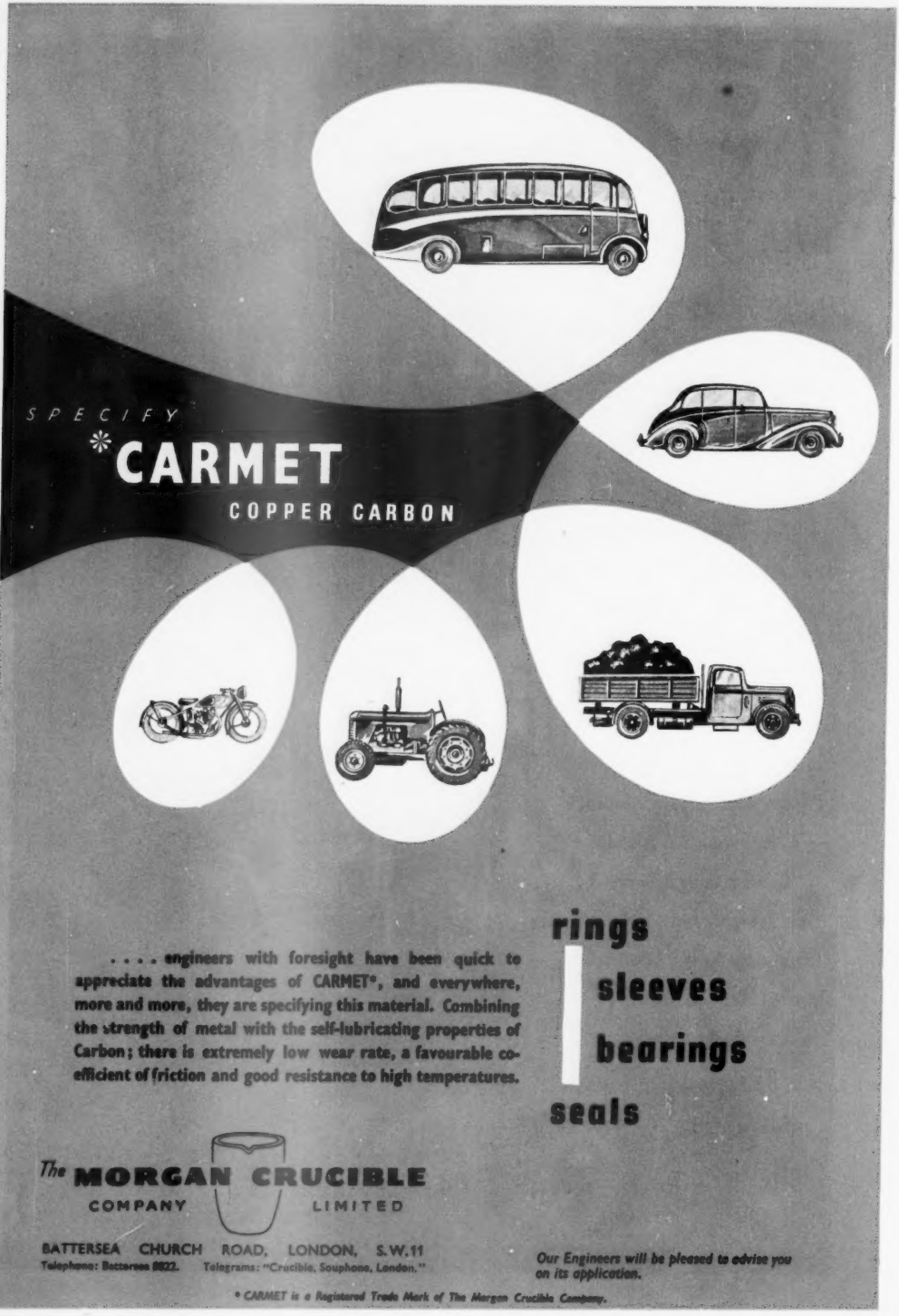
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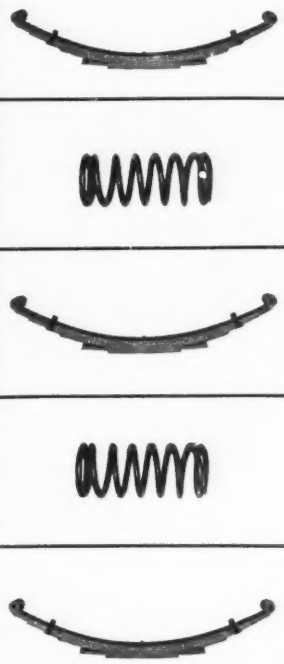

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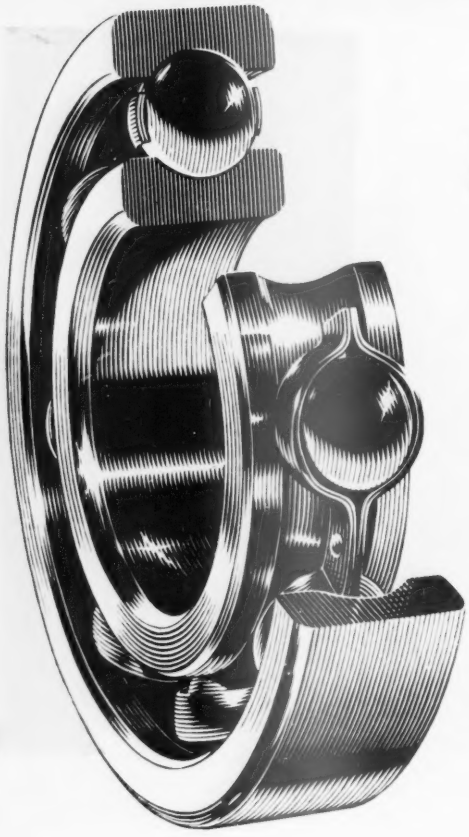

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Design, Materials, Production Methods, and Works Equipment

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Commercial Vehicles

DURING these post-war years, the commercial motor industry has had an export record that is a matter for justifiable pride, and the recently concluded biennial exhibition suggests that this record will be maintained. Those engaged in the industry realize only too well that the days of a seller's market are over, but nonetheless there is still quiet confidence that markets will be retained.

It is really a healthy sign that the first impression gained at the exhibition was that power units, transmissions and chassis generally showed but little change from those exhibited two years ago. This could be, but is not, a sign of stagnation. It is rather a proof that throughout the post-war years British design of commercial vehicles has been sound, so that while from year to year there are detail improvements, there is little possibility, and probably even less need, of any sudden marked change.

In a buyer's market first cost is always important, and manufacturers have done all that is possible to reduce prices while at the same time maintaining a high standard of quality. But even more important than first cost is the question of operating costs, and the modifications that have been made are generally such as will reduce these costs. In brief, it may be said that the general tendency is towards lighter constructions in all parts of the vehicles.

Engines and transmissions

Power units and transmissions generally show but little apparent change. For a given capacity the new engines show an increase in power output in conjunction with a reduction in fuel consumption, but this is due to refinements and not to any radical changes in design. This applies to both petrol and diesel engines.

Transmissions follow conventional lines, most of the heavy vehicles having five-speed gearboxes, although some makers have developed six-speed boxes. Not a single vehicle was fitted with automatic transmission. This caused some comment, but it is really evidence of the good sense of British designers rather than a sign of innate conservatism. In the present stage of development there is good reason for doubting whether the incorporation of any of the known forms of automatic transmission can be economically justified. What little advantage can be gained is apparently much more than counterbalanced by the increase in first cost and in fuel costs. For the time being it does appear to be sound policy to refrain from fitting such devices.

Bodywork

Without any doubt the most marked differences between vehicles shown in 1950 and those shown recently were in the bodywork. Overseas demands have caused the introduction of some measure of unconventionality into the products of British body-building plants. Several interesting bodies were exhibited, and the necessity for coping with extreme climatic conditions has been responsible for the development of bodywork that is very different from that demanded by operators in this country. There is, however, one aspect that is common to nearly all bodies; that is the use of metal in place of wood for structural members.

This, of course, is not a recent development. Ever since the war there has been a tendency to use a higher proportion of light alloys in body construction. The consequent saving in weight is a desirable feature in any road vehicle, and more particularly where operating costs must be closely controlled. Experience gained in the handling of light alloys for aircraft construction has been applied to bus and coach body building, and a well-worth-while saving in weight has been achieved without in any way impairing the strength of the structure.

Another matter that has received close attention is that of unit construction. This is a method that facilitates production and reduces costs. From the operator's point of view this construction is advantageous, since it ensures that a damaged vehicle can be returned to service with a minimum of delay, jig produced units being readily available. This is particularly important to overseas operators who can prepare for such eventualities by stocking replacement units. With this in mind, designers have paid special attention to panel and body member attachment.

Body builders have also taken full advantage of the additional floor space made available by the adoption of under-floor engines. In this connection, however, it is felt that body engineers would be well advised to pay greater attention to the matter of insulation against noise and vibration. There is a wide selection of sound-deadening materials and many forms of appliances to reduce the effects of vibration. Private car technicians have made full use of these devices in their quest for silence. Public service vehicle engineers should do the same. It would be unfortunate if the further development of the under-floor

engine were retarded in any way by faults that are readily capable of solution.

To sum up, it may be said that although there are no outstanding changes from accepted practice, there is much evidence of steady progress. To recapitulate, the aim of all the makers has been to produce vehicles that will give reduced operating costs. This includes not only reduced running costs, but servicing costs also. In fact, one of the outstanding features of the new designs is the care that has been taken to make servicing and repairs as easy as possible. A full review of the exhibition will be published in the November issue of this journal.

Machines and Productivity

THE aim of the machine tool manufacturer is to make possible greater production, of higher quality and at a lower cost in terms of both money and manpower. In this, a very great measure of success has been achieved. Some of the rates of metal removal that are now employed would have been thought fantastic only a few years ago. This is in part due to developments in cutting materials, which have been matched by machine developments.

So far as the automobile industry is concerned, it is debatable whether any further developments to give higher rates of metal removal would be worth while. The machines would be considerably more expensive and the returns would hardly be commensurate. Most machining operations in the automobile industry have a short time cycle in which loading, unloading and fixture cleaning already occupy a considerable proportion of the total time. In the well-equipped motor factories these idle time factors have been reduced to an almost irreducible minimum and any increase in speeds and feeds would not have a proportionate effect on the production rate.

Of much greater importance to the industry is a greatly increased tool life. Therefore the makers of cutting materials should concentrate on developing tools which will last longer between regrinds rather than tools that will cut faster. The machine tool makers have played their part by developing machines that are free from vibration and chatter, two of the most destructive factors in tool life. In fact, freedom from vibration was an outstanding feature of many of the high speed machines exhibited at Olympia.

There was evidence, however, of one tendency that is not altogether to be welcomed; that is, the tendency for machines to become more and more complex and complicated. This is in part the fault of the user who will often specify a type of machine that will perform functions, some of which will never be required during the life of the machine. On the other hand, the machine tool industry is not free from blame in this matter.

For years there has been a definite tendency to reduce to a minimum reliance on the skill of the operator in machining operations. Within limits this is sound policy but in some cases it would appear that the limits have been exceeded, and costly complications are incorporated in the machine for a relatively small advantage. Furthermore, these complications can and do entail the use of delicate mechanisms, which because of their delicacy are subject to troubles and failures which do not affect other parts of the machine, but which cause the machine to be idle until the trouble is rectified. In many cases they also call for specialist attention.

The best machine for any given job is the simplest that will perform the job efficiently. This may sound like stressing the obvious, but like many other obvious things it is often disregarded. It is because we believe in simple solutions wherever they are possible that we have doubts about the advisability of some of the very elaborate transfer machines that have recently been installed in some factories.

The relatively simple straightforward type of transfer machines used by Vauxhall Motors Ltd. for machining Bedford engine cylinder heads has much to recommend it when the volume of production is sufficiently high, but whether a machine with fifteen or twenty stations is warranted is open to doubt. For our part, we incline to the view that the method of grouping three or four unit heads together to form in effect a single machine is generally a better solution. The first costs for machines will be much lower than they would be for multi-station transfer machines; this in itself is important in that amortization charges will be much lower. In addition, the machines are basically simpler and less prone to breakdowns, and if breakdown does occur, the transfer machine is not so easily or readily brought into service again. Nor has the very elaborate type of transfer machine the same production flexibility as the simpler machines. In fact, it is possible to envisage transfer machines acting as a brake on progress in product design. To set against this there is, of course, the reduction in the labour force necessary to produce a given number of components, but this is often considerably exaggerated.

CONTENTS

Page

Editorials 369-370
Commercial Vehicles. Machines and Productivity

The New Austin "7" 371-385
A Well-equipped Car of Light Weight

Centrifugal Filtration 386
An Interesting Development by Glacier Metal Co. Ltd.

Engine Lubricating Oils. By J. G. Withers ... 387-393
The Performance of Premium and Heavy-duty Qualities

Recent Publications 394
Brief Reviews of Current Technical Books

Page

Machine Tools 395-404
A Review of Trends Disclosed at the Recent Exhibition

Tappet Wear 405-408
Some Development Work Leading to the Adoption of Cast Chromium Alloy Tips

Starting in Extreme Cold. By J. D. Richards 409-413
A Study of the Special Difficulties with Small-capacity Engines

Tyre Curing 414
Mechanized Equipment for Faster Production

THE NEW AUSTIN '7'

A Well-equipped Car of Light Weight

WHEN the new Austin '7' was first planned, the general trend in design was towards more and more refinement for both interior appointments and general mechanical features. Although there was obviously a large market for a small inexpensive car, it was not considered wise to produce a vehicle on the very austere lines of the Continental economy cars. This model is definitely in the small car class, but nevertheless it is a full four-seater with an enclosed luggage boot of reasonable size. Unlike most of the economy cars, it has Dunlopillo upholstery, balanced-lift windows, and provision is made for a Smiths recirculator-type heater, if required.

Although these amenities are provided, the basic price of the car is remarkably low. There is a fairly close price per lb-weight ratio for all quantity produced cars, and there is little doubt that the lightweight structure of this vehicle has much to do with its low cost. The kerb weight of the car is about 1475 lb (no fuel), the dry weight of the engine is 219 lb, and that of the gearbox and clutch is 43 lb. In the white, the body-chassis structure weighs only 410 lb.

This light structure weight has not been obtained by any sacrifice in strength. In fact, the car is considerably stiffer and stronger than would have been possible if it were constructed with a conventional separate chassis. When the torsional stiffness was measured by a method similar to that recommended by M.I.R.A., the deflections were too small to give accurate measurable figures. A very severe test programme has been carried out to prove the vehicle to be thoroughly road worthy. In addition to many thousands of miles running in this country, the tests included a 5,000 miles run in European countries, over both mountainous and dusty terrain, as well as pavé and other bad roads. Moreover, 1,500 miles were covered on the pavé track at the M.I.R.A. testing ground, modifications to strengthen any weak points being made as found necessary during the test.

In the design of a small car, the greatest ingenuity must be exercised in order to fit all the mechanical components into the vehicle, while still maintaining the requisite standard of passenger comfort. Indeed, to design a satisfactory four-door saloon with a wheelbase of only 6 ft 7½ in is no mean feat. Contemporary models with which the Austin '7' may be compared are the Morris Minor, whose wheelbase is 7 ft 2 in, and the Renault 760 saloon with a wheelbase of 6 ft 10½ in. In this type of car it is particularly difficult to reduce the height in the same proportion as the other dimensions, so that the lines shall be as pleasing as those of a larger car. The result is that there is a tendency for the smaller vehicles to appear too high. This tendency has been overcome by the use of a chassisless construction which allows the floor to be kept at the lowest possible level.

SPECIFICATION

ENGINE: Four cylinders. Bore and stroke 58mm × 76mm. Swept volume 800cm³. Maximum b.h.p. 30 at 4,800 r.p.m. Maximum b.m.e.p. and torque respectively 120lb/in², and 40lb-ft at 2,400 r.p.m. Compression ratio 7·2: 1. Three bearing crankshaft, partly balanced. Overhead valves, push rod operated. Zenith downdraught carburettor type 26 JS with 18mm choke.

CLUTCH: Borg and Beck single dry plate, 6½in diameter.

GEARBOX: Four forward speeds and one reverse, synchromesh on 2nd, 3rd and top. Ratios: top 1: 1, 3rd 1·68: 1, 2nd 2·59: 1, 1st 4·09: 1, reverse 5·18: 1.

REAR AXLE: Hypoid: three-quarter floating, with banjo casing. Ratio 5·14: 1.

REAR SUSPENSION: Semi-elliptic reverse camber leaf springs. Armstrong double-acting lever-operated shock absorbers coupled by an anti-roll bar.

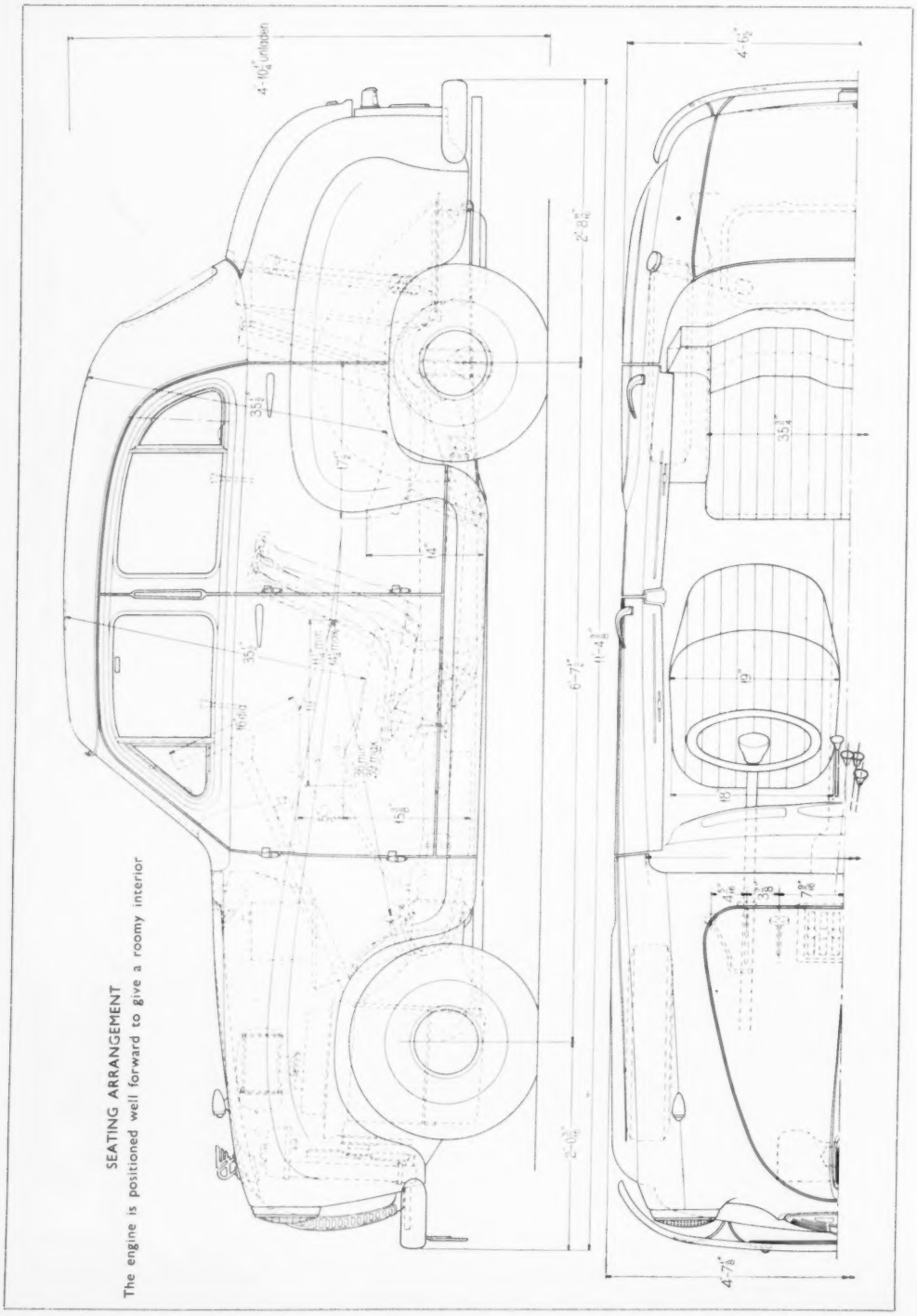
FRONT SUSPENSION: Independent, coil springs, with lower wishbones, and single upper transverse links mounted on Armstrong double-acting shock absorbers.

BRAKES: Front: Lockheed hydraulic, 2LS. Rear: Lockheed mechanical, trailing shoe type, operated by separate hydraulic cylinder. All brake drums 7in diameter × 1¼in wide.

TYRES: Dunlop 5.20 × 13.00. Pressures at full load 20lb/in² front, 23lb/in² rear.

DIMENSIONS: Wheelbase 6ft 7½in. Track front 3ft 9¼in, rear 3ft 8¾in. Dry weight 1456lb.

The Austin '7' saloon



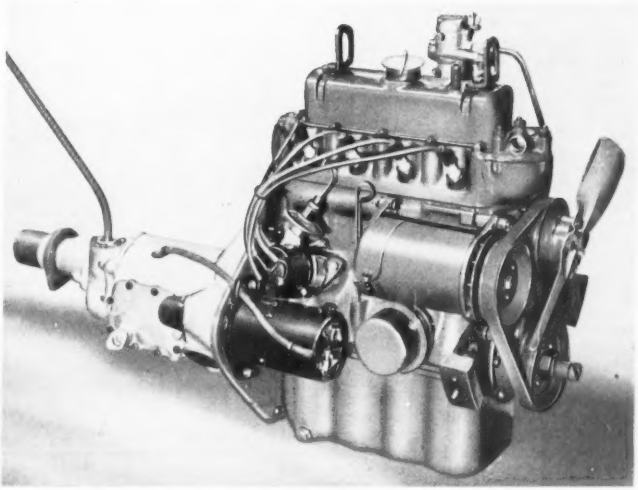
SEATING ARRANGEMENT

The engine is positioned well forward to give a roomy interior

It necessitates having a rather high propeller shaft tunnel, but this is completely unavoidable and of no disadvantage.

As can be seen from the illustration showing the interior arrangement, the greatest possible overall length in the saloon has been obtained in two ways. The first is by positioning the engine well forward, at an angle of 2½ deg from the horizontal longitudinal axis of the car, while the second is the setting of the rear seats as far back as possible. There is a limit to the advantage that can be gained from the forward engine position. This limit is imposed by the front wheel arches which, as the toe-board is progressively moved farther forward, intrude more and more inwards and reduce the space for the pedals. In this car, a good compromise has been reached, and the space between the toe-board and the clutch bell-housing has been taken up by the track rod.

With regard to the back seats, restrictions are imposed on their possible rearward position principally by the need to accommodate, at full bump, the casing of the hypoid axle. In so small a car only two passengers can be carried in the rear seats. It is therefore practicable to press the centre portion of the seat pan between the two passengers, to provide clearance for both the propeller shaft and part of the nose-piece of the axle. Another limitation is the width obtainable between the rear wheel arches. However, in view of the desirability of limiting the amount of load that can be imposed on this type of vehicle, this is not serious provided that two passengers can be comfortably accommodated. Additional width has been gained where it is most required by sloping the inner walls of the wheel arches outwards towards their base.



Complete power unit, right side. The spark plug bosses are recessed into the cylinder head

Engine

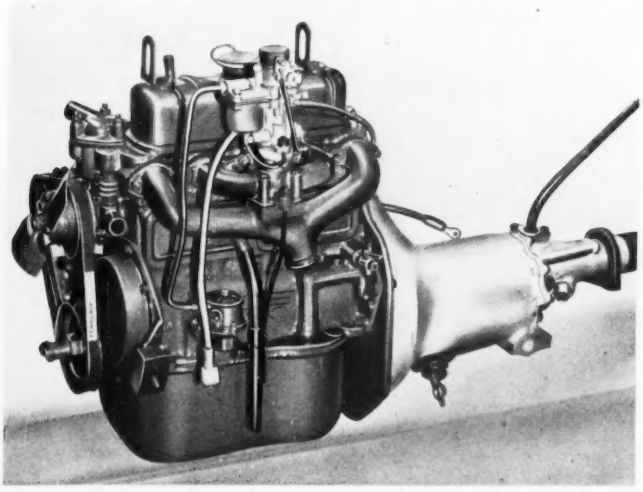
The 800 cc engine develops 30 b.h.p. Its stroke/bore ratio is 1·31 : 1. By modern standards this is rather a high figure, but any attempt to reduce the stroke and increase the bore would, of course, have added to the overall length of the unit. Moreover, this ratio has given satisfaction for many years past, in the A.40 for instance, and the Company are loth to depart from it. In any case the stroke is only 3 in and the piston speed is low. The connecting rod length/stroke ratio is 1·935 : 1. At 4,800 r.p.m. when maximum b.h.p. is developed, the mean piston speed is just under 2,400 ft/min. The maximum b.m.e.p. is 120 lb/in^2, and maximum torque is 40 lb/ft, both values being obtained at 2,400 r.p.m. In terms of b.h.p./in^2 piston area, the performance is good for this class of car, the figure being 1·84 h.p/in^2.

In the design of an engine for a small car, economy in both weight and overall dimensions is of utmost importance. Although the reasons for restricted overall dimensions are apparent, it is not quite so obvious why in this class of vehicle, weight is more important than in a larger one. The reason is that the load the car must carry, that is to say four passengers and luggage, is not scaled down in proportion to its size. Moreover, many standard components, such as electrical equipment, used in larger cars must be employed. Therefore, if the power/weight ratio is to be adequate, it is essential that the weight is reduced to a minimum in the engine, as well as in the other mechanical components and the structure.

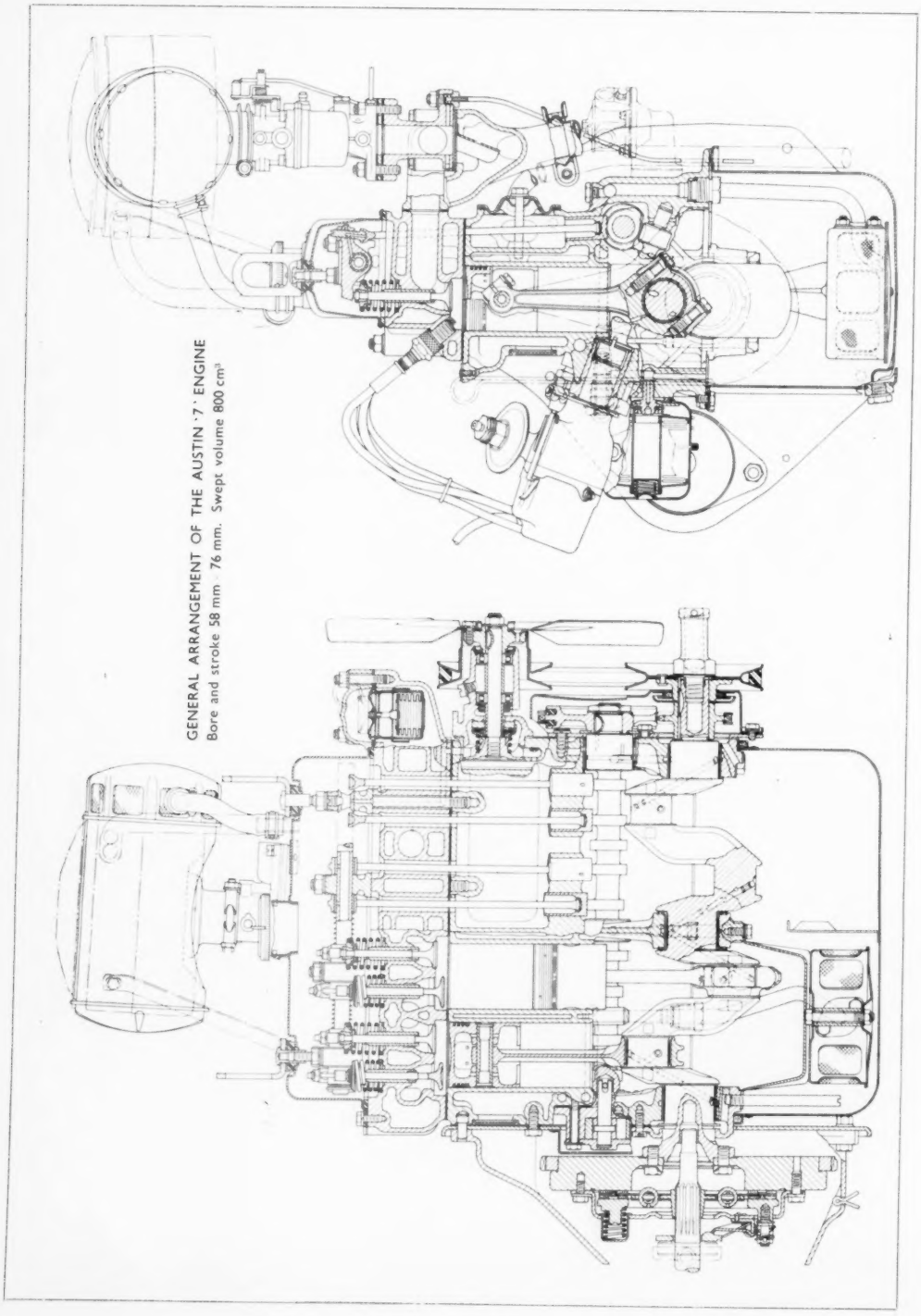
As far as general layout is concerned, the engine is fairly conventional, the most unusual feature being the position of the oil pump, which is driven directly from the rear end of the camshaft. A dished pressed steel cover is bolted to the aperture in the clutch bell housing closing plate through which the pump projects about ¼ in.

The ignition contact breaker and distributor unit is mounted in a boss on the right-hand side of the crankcase, between Nos. 3 and 4 cylinders. It is driven from an integral spiral gear on the camshaft, which is on the left-hand side. Below the distributor is the starter motor, in front of which is the oil filter. The dynamo, which is also on the right-hand side, and the fan and water pump unit bolted to the front of the engine, are driven by the usual triangular vee-belt arrangement.

On the left-hand side of the engine are the inlet and exhaust manifolds,



A thin-plate hot-spot is clamped between the inlet and exhaust manifolds



GENERAL ARRANGEMENT OF THE AUSTIN '7' ENGINE
Bore and stroke 58 mm. × 76 mm. Swept volume 800 cm³

and the fuel pump which is actuated by an eccentric on the camshaft between Nos. 1 and 2 cylinders. A crankcase ventilation pipe is welded to the tappet cover, and the ventilation system is completed by another pipe between the rocker cover and the air cleaner and silencer. On each side of the front engine plate, flanges are turned forward to carry a sandwich-type rubber mounting.

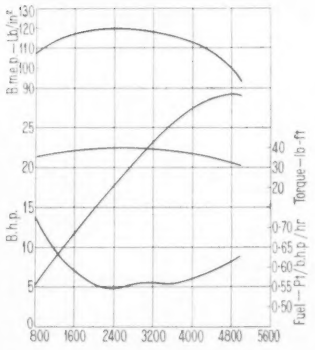
The rear mounting is a rubber ring of U-section, with its concave portion facing inwards. A collar cast on the cylindrical rear extension of the gearbox engages in the rubber ring, which is housed in a channel-section, spot welded round the inside of the propeller shaft tunnel. Assembly of the engine and gearbox unit with the rubber ring in position is from below. The unit is held up by a pressed steel bridge-piece pulled up to the floor below the channel by two bolts. This method of mounting gives adequate fore and aft location to the power unit, while at the same time absorbing torsional vibrations. However, it must be rather stiff so far as vertical loading is concerned, although no doubt, since the rear mounting is positioned well behind the engine, vertical vibratory loads are small. The design of engine mountings is always something of a compromise, since undue flexibility can be the cause of clutch judder.

Cylinder block and crankcase

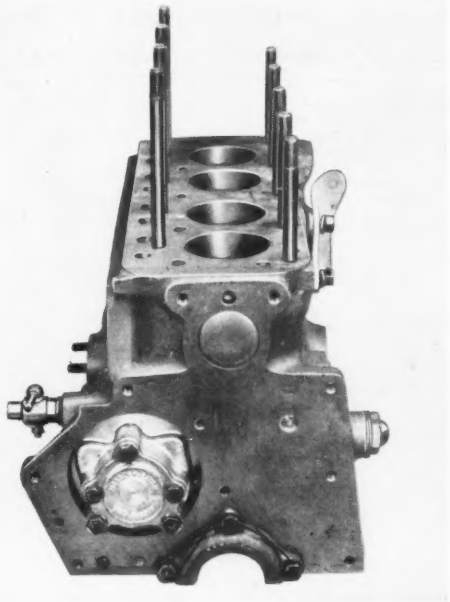
An integral cylinder block and crankcase, of B.S.S.1452 grade 17 cast iron, is employed. The distance between the axes of the centre pair of cylinders is $3\frac{1}{4}$ in, while that between Nos. 1 and 2, as well as 3 and 4, is $2\frac{25}{32}$ in. With a wall thickness of $\frac{3}{16}$ in, this gives a minimum water jacket space between the cylinders of about $\frac{21}{64}$ in. The unjacketed length at the skirt of the cylinder is approximately $\frac{5}{8}$ in, except on the right-hand side where it is locally $1\frac{1}{2}$ in to clear the oil gallery.

The rear face of the crankcase is machined to carry the integral gearbox and clutch casing and the $\frac{1}{8}$ in thick closing plate which is flanged to increase its stiffness. Positive location is given by two dowels, and the unit is secured by six set-bolts. Four bolts and nuts below the crankcase are used to pull the closing plate up to the clutch housing. The oil pump is recessed into the rear wall immediately behind the camshaft.

At the front of the engine, the water pump is spigot-mounted into the block, so that the whole of the pump rotor is positioned behind the front wall. This reduces the overall length of the unit by about $\frac{13}{16}$ in, as compared with the more usual layout that has the pump bolted on forward of the front face of the block. Below the water pump, the front engine plate and the pressed steel timing gear cover are secured by nine set-bolts, locked by spring washers. A Langite sealing washer is employed. The crankcase walls extend down to the level of the axis of the three main journal bearings. These are carried one in each end wall, and one in a transverse ribbed web. The bearing caps are secured by $\frac{7}{16}$ in. diameter En 16 set-bolts locked with tab washers. Positive location is effected by dowel tubes round the bolts.



Engine performance curves. No fan or dynamo fitted, shop exhaust system employed



Driven from the camshaft, the oil pump is recessed into the crankcase rear wall

Crankshaft, connecting rod and pistons

Vandervell D2 bimetal bearing shells are employed for the main bearings carrying the En 16 forged crankshaft. Axial location is effected at the centre bearing by Vandervell D2 bimetal washers located in recesses in the front and rear face of the web and cap. These thrust washers are divided horizontally. Each is prevented from rotating by a tab on the bottom half that engages in a slot in the face of the bearing cap. All the main journal bearings are $1\frac{3}{4}$ in diameter $\times 1\frac{9}{16}$ in long, and the length of the crankshaft between the outer faces of the front and rear webs is $10\frac{23}{32}$ in. Until 1936 when a three bearing crankshaft was introduced, the early Austin '7' engines, which had no centre bearing, had unbalanced crankshafts. However, in this model, partial balancing is effected by counterweights forged integrally with each vertical crankweb but not machined. Without these weights, the unbalanced loads on the intermediate bearing would be too heavy.

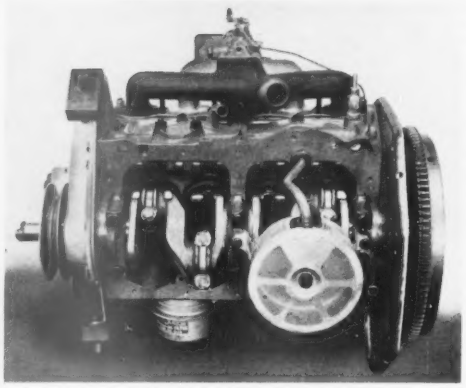
Immediately behind the rear bearing is an integral oil thrower ring, and behind this, machined on the shaft is a $2\frac{5}{8}$ in diameter scroll-type oil return. Round the oil return is a housing, formed in the rear bearing cap and completed by a separate cast-iron semi-circular piece bolted to the rear face of the crankcase. A duct is drilled through the bearing cap, and in it is fitted a baffle tube to drain oil away from the space around the thrower ring. The tube extends to below the oil level in the sump so that there is no possibility that momentary high pressures in the crankcase will blow oil vapour into the clutch housing. Four $\frac{3}{8}$ in diameter set-bolts locked by tab washers, secure the $9\frac{1}{2}$ in diameter flywheel, of 18-ton cast-iron, to a flange on the crankshaft. A shrunk-on starter ring gear is fitted. It is made of a mild steel having a carbon content of 0·2-0·5 per cent with 6 per cent manganese. The tail end of the crankshaft is counterbored to house the Oilite bush for the clutch main shaft.

Forged H-section connecting rods made of En 16 are employed. They have a centre-to-centre length of $5\frac{1}{4}$ in. The vertical axes of the connecting rods are offset in relation to the central vertical planes of the crankpins; No. 1 rod $\frac{1}{8}$ in to the front, No. 4 rod $\frac{1}{8}$ in to the rear, and Nos. 2 and 3 $\frac{1}{8}$ in towards the centre bearing. In order that the rods may be withdrawn through the cylinder bores, the big ends are divided at an angle of 50 deg to the axis of the rod. Vandervell D2 bimetal bear-

ing shells $\frac{7}{8}$ in long × 1 $\frac{13}{32}$ in diameter are fitted. They are located by the usual pressed-out projections at their abutting faces, engaging in grooves machined in the caps and the rods. The bearings caps are secured by $\frac{3}{8}$ in diameter En 16 set-bolts locked by tab washers. These bolts are reduced to $\frac{5}{16}$ in diameter for a length of $\frac{7}{16}$ in so that the stress is as uniform as possible along their length. This improves their fatigue resistance. Lateral location of the cap is effected by machined shoulders that register against shoulders on the connecting rod.

The En 34 gudgeon pin is clamped in the small end by a $\frac{5}{16}$ in diameter En 16 set-bolt locked by a spring washer. Its outside diameter is $\frac{9}{16}$ in; it is drilled out $\frac{3}{8}$ in diameter for a length of $\frac{7}{16}$ in at each end, while the remaining $1\frac{1}{8}$ in is drilled $\frac{5}{16}$ in diameter. The effective bearing length in each piston boss is $\frac{3}{4}$ in. Vertical holes drilled in the bosses assist splash lubrication.

T-slotter Alumilated Lo-Ex pistons are employed, the slots being on the side opposite to the thrust face. The castings for these are supplied by Wellworthy Ltd. and the British Piston Ring Co. Ltd. A lip round the inside of the lower rim of the skirt ensures adequate stiffness. The crowns of the pistons have a machined concavity that takes the form of a frustum of a sphere, the chord diameter of which is about $1\frac{3}{4}$ in. There are three plain rectangular - section compression rings and one oil control ring. All four are fitted above the gudgeon pin. The face width of each of the compression rings is 0·070 in and the radial thickness, which is given by the formula $\frac{d}{28}$ where d is the cylinder bore, is 0·08 in. The slotted oil control ring which has the same radial thickness, has a face width of 0·125 in.



The crankshaft is partly balanced to relieve the loads on the centre bearing

CAMSHAFT PERFORMANCE DATA AT 4,800 ENGINE R.P.M.

Maximum tappet positive acceleration (flank)	7300 ft sec²
Maximum tappet negative acceleration (nose)	3420 ft sec²
Maximum tappet velocity	7·35ft sec
Lift at camshaft	0·225 in
Nominal period of cam	115 deg

Timing gear, camshaft and valve gear

A $\frac{1}{2}$ in pitch single-strand chain, supplied by Renold and Coventry Chain Co. Ltd., is used for the timing gear drive. A 1 in diameter forward extension of the crankshaft carries the B.S.S.1452 grade 14 driving sprocket. In front of the sprocket is a flanged boss to which is riveted the two-piece pressed steel fan-belt pulley. To the rear of the pulley and round its boss is a felt ring housed between the inner face of the 20 s.w.g. pressed steel timing cover and a spot welded-on dished washer. An L-section ring is pressed on to the outer periphery of the dished portion in such a way as to prevent any oil that may drain down the front wall of the timing cover from running over the rear face of the washer and directly on to the felt seal. The whole assembly is completely shrouded by a dished oil thrower ring clamped between the pulley boss and the timing gear sprocket.

Both the sprocket and the pulley boss are driven by a single Woodruff key. They are pulled up against a shoulder formed by the crankshaft front journal bearing, by a $\frac{3}{8}$ in diameter bolt with a special head. This bolt is screwed into the front end of the crankshaft. Its head is extended forward to incorporate the dogs for the starter handle. The B.S.S.1452 grade 14 half-speed wheel is keyed on to a $\frac{7}{8}$ in diameter extension at the front of the camshaft, and secured by a nut locked by a tab washer. The rear end of the half-speed wheel boss is shouldered to form a groove between it and the front face of the adjacent camshaft bearing. A Vandervell D2 bimetal thrust plate $\frac{3}{32}$ in thick registers in the groove to locate the camshaft axially. This plate is secured by three set-bolts passing through the engine plate and screwed into the front wall of the crankcase.

In the front crankcase wall, a Vandervell D 2 bimetal bush is pressed into the boss for the camshaft bearing. Neither of the other two bearings is bushed. The bearing diameters are: front 1 $\frac{13}{16}$ in, centre $1\frac{3}{8}$ in, rear $1\frac{1}{8}$ in; the shaft diameter between the bearings is $\frac{13}{16}$ in. In the tail end of the En 34 camshaft, a $\frac{3}{16}$ in diameter axial hole carries the short driving spindle for the oil pump. A $\frac{1}{4}$ in diameter peg with flats machined, one on each side of its periphery, is pressed into a hole drilled diametrically through the camshaft, and engages the slotted end of this spindle. The eccentric for the fuel pump is between Nos. 1 and 2 cylinders. Between Nos. 3 and 4 cylinders there is a spiral gear to drive the contact breaker and distributor. As in all Austin engines, the cams, which incorporate variable eccentricity ramps, are designed broadly on the principles described in the paper read by J. L. H. Bishop, A.M.I.Mech.E., before the Institution of Mechanical Engineers in 1951 and reproduced in the June issue of *Automobile Engineer*.

Chill - cast piston - type tappets with oil drain holes, operate in bosses in the crankcase. The effective length of the $\frac{1}{4}$ in diameter En 15 push rods is about $8\frac{1}{4}$ in. An upset spherical seating at the bottom of each rod is supported in the tappet; at the top a cup end is formed to seat the induction hardened En12 tappet adjustment screw, which is carried in the rocker end and secured by a lock nut. Both ends of the rods are induction hardened. The En 12 rockers have Clevite bushes pressed into their bores. Their end pads, which bear on the top of the valve stems, are also induction hardened. A rocker ratio of 1·25 : 1 has been adopted.

The induction hardened hollow rocker shaft of En 12, has an outside diameter of $\frac{9}{16}$ in, and an inside diameter of $\frac{5}{16}$ in. It is sealed at one end by a pressed-in plug and by a screwed plug at the other, in order that it may be cleaned during service. The shaft is mounted on four die-cast S.A.E.306 H.T. aluminium pedestals, each of which is bored to carry it. On assembly, the front pedestal is placed in position first, and then located by a conical ended $\frac{1}{8}$ in diameter grub screw, inserted from the top to engage in a tapered hole in the shaft. The remaining components, that is to say the pedestals, rockers, and their separating springs are then assembled on the shaft from the rear. They are retained by a double coil spring washer and a large diameter plain washer, held in position by a split pin through the extreme rear end of the shaft. A similar arrangement is employed to retain the front rocker on the shaft.

A single spring is fitted to each of the vertically positioned valves. At its lower end it seats in a recess machined in the cylinder head; at its upper end it bears against a pressed steel retainer for a rubber washer. Above this is the usual valve retaining washer, held in position by a split tapered En 14 collet. A hairpin clip in a groove round

each collet ensures that should the spring break, the valve cannot fall into the cylinder.

The inlet valve guides are $1\frac{7}{8}$ in long and have plain bores. Those for the exhaust valves are also $1\frac{7}{8}$ in long but for a length of $\frac{1}{32}$ in at the lower end of their bores there is $\frac{1}{32}$ in clearance round the shouldered valve stem. This prevents the build up of carbon deposit on the stem. Despite the curvature of the port, the lower end of the exhaust valve guide boss is square to its axis. This leads to more accurate machining than would otherwise be possible.

VALVE DATA

	Inlet	Exhaust
Material	Silchrome	XB steel
Head diameter	$1\frac{3}{32}$ in	1 in
Throat diameter	1 in	$\frac{7}{8}$ in
Stem diameter	0·2812 in	0·2795 in
Spring rate	114 lb in	
Spring length free	$1\frac{5}{8}$ in	
Spring length installed	$1\frac{13}{16}$ in	
Spring frequency	26,300 c.p.m.	
Number of coils	$4\frac{1}{2}$	
Coil diameter	$1\frac{7}{64}$ in	
Valve lift	0·285 in	
Rocker ratio	1·25 : 1	
Valve crash speed	5750 r.p.m.	
Tappet clearance	0·015 in	
Valve opens	5 deg B.T.D.C.	40 deg B.B.D.C.
Valve closes	15 deg A.B.D.C.	10 deg A.T.D.C.
Ignition timing	T.D.C. fully retarded	

Cylinder head, manifolds and carburettor

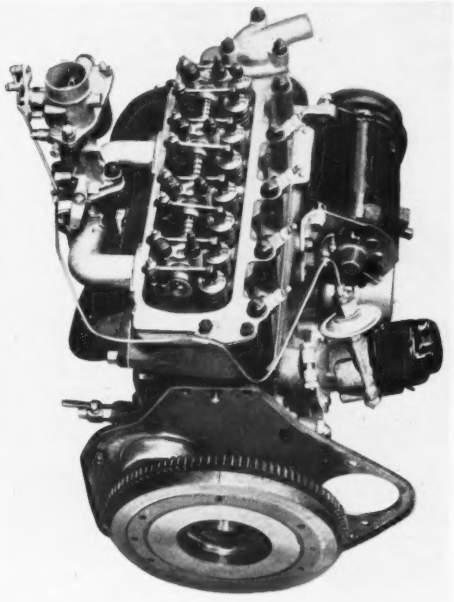
A cast-iron cylinder head is employed. The depth of the casting is $2\frac{1}{4}$ in and the overall width $5\frac{1}{4}$ in. Some weight has ben saved by reducing the width of the head above the spark plug bosses to $4\frac{1}{8}$ in. A forward extension of the casting houses the thermostat, and at the rear, provision is made for a bolted-on adaptor for a heater connection. When a heater is not installed, a blanking plate is fitted.

A varnished steel gasket is used for the cylinder head joint and a washer made of Langite reinforced with aluminium for the rocker cover joint. The rocker cover, on which an oil filler tube is welded, is held down by two special insulated nuts fitted to extra long pedestal holding down studs of En 16, one at each end of the rocker assembly.

Combustion chambers of Weslake design are incorporated. They are, like those of the other Austin engines, flat topped and roughly heart shaped in plan. The theory of these combustion chambers was described in "Power Output" in the August issue of *Automobile Engineer*. Effective cooling is ensured by having the spark plug bosses as well as the inlet and exhaust ports and guides, completely surrounded by coolant.

The valve seats have a face angle of 45 deg and are cut directly in the casting. Both the inlet and the exhaust valve ports pass horizontally through the head after being turned through an angle of 90 deg from the seats. The radius through which the axes of both ports are turned is $\frac{33}{32}$ in. A double Compo joint washer is fitted between the manifolds and the cylinder. The exhaust manifold is of Qualcast cast-iron, and the inlet is of die-cast aluminium. A 16 s.w.g. plate with an exhaust gas deflector welded to its lower face forms a hot spot at the junction between the two manifolds, which are bolted together. An adaptor for a manifold drain pipe is screwed into the casting immediately above this plate. The pipe is extended down to $\frac{1}{2}$ in below the level of the sump face joint.

A 26 JS downdraught Zenith carburettor is fitted. It has an 18 mm choke tube. The jet sizes are: main jet 95, main air jet 160, slow running jet 40, slow running air bleed 80, and needle and seat 1·5 mm. Fuel is drawn from the $5\frac{1}{4}$ gallon tank by an A.C. Sphinx mechanical Y-type pump, which incorporates a gauze strainer. An A.C. air filter and intake silencer is employed.



Arrangement of overhead valve gear. Hairpin clips retain the collets to support the valves in the event of spring breakage

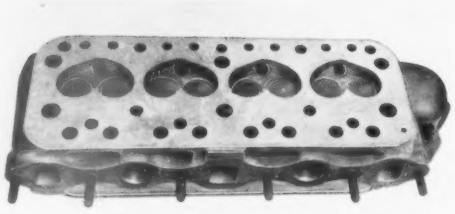
Water pump and cooling system

A cord-reinforced rubber vee belt $\frac{3}{8}$ in wide × $\frac{7}{16}$ in deep with an included angle of 40 deg, drives the water pump at 1·2 times crankshaft speed. The B.S.S.1452 grade 12 cast-iron water pump body is spigoted into the front face of the cylinder block. It is secured by four set-bolts locked by spring washers. The En 16 pump spindle is carried in two ball bearings separated by a distance tube and housed in a forward extension of the body. Provision is made for lubricating the bearings through a duct sealed by a screwed-in plug. The boss for the fan and pulley is fitted on the front end of the spindle with the drive taken through a Woodruff key. A steel shroud washer round the rear end of the boss protects the bearing. It is retained by a circlip in the pump housing.

The outer race of the front bearing has no positive axial location in the housing, but its inner race is clamped between the rear end of the boss and the distance tube. Moreover, the outer race of the rear bearing is not positively located against movement in the forward direction. Between its rear face and a shoulder in the housing is a pressed steel annulus carrying a felt seal. This seal is mounted round a distance tube between the inner race of the bearing and the $2\frac{1}{8}$ in diameter cast iron rotor which is screwed on to the rear end of the spindle and locked by peening. The restraining force, holding the outer race of the rear bearing and the felt seal against the shoulder, is provided by the spring of the water seal.

Although the water seal is of a similar design in all the Austin range of engines, it is somewhat unusual. The coil spring is made of phosphor bronze. At its rear end it bears against the rotor, and at the front against a brass cup-shaped retainer for the rubber seal. The retainer, which is positioned round the boss of the rotor, has four projections registering in slots machined in the boss. In front of the retainer is a moulded rubber disc round which there is a bonded-on cupped washer. This washer is clamped between the distance tube and the rotor boss. The front end of the distance tube bears against the inner race of the rear ball bearing. A rim formed round the periphery of the rubber

sealing disc is held by the brass retainer against a thrust washer made either of Durestos or similar material. The front of this washer bears against a ground face in the pump casing. Two lugs on the periphery of the thrust washer register in slots in the brass retainer, so that the oil seal assembly rotates with the rotor and its spindle. There is no thrower ring between this seal and the ball races, but there is the usual drainage space.



The high efficiency combustion chambers are heart-shaped in section

Coolant passes from the pump round the cylinder walls and through ducts into the cylinder head. The thermostat valve is clamped, together with a Velumoid joint washer, between the head and the aluminium water outlet casting, the whole assembly being secured by three studs and nuts. A fin and tube radiator of 190 in² frontal area and 1¼ in block depth is employed. The water filler cap is of a patented design employed on all Austin vehicles, with the boss, into which the filler cap is secured by a bayonet type fitting, recessed below the level of the top of the header tank. Thus, it is impossible to over-fill the tank, and there is always adequate air space above the coolant to allow for expansion. The overflow pipe is taken, inside the tank, from a dome on top, and discharges into the recess at a point near the boss for the filler cap. With this arrangement, mainly steam and air pass through the overflow pipe to relieve any pressure that is built up, so that little or no coolant is lost. The capacity of the coolant system is 8½ pints.

Oil pump and lubrication system

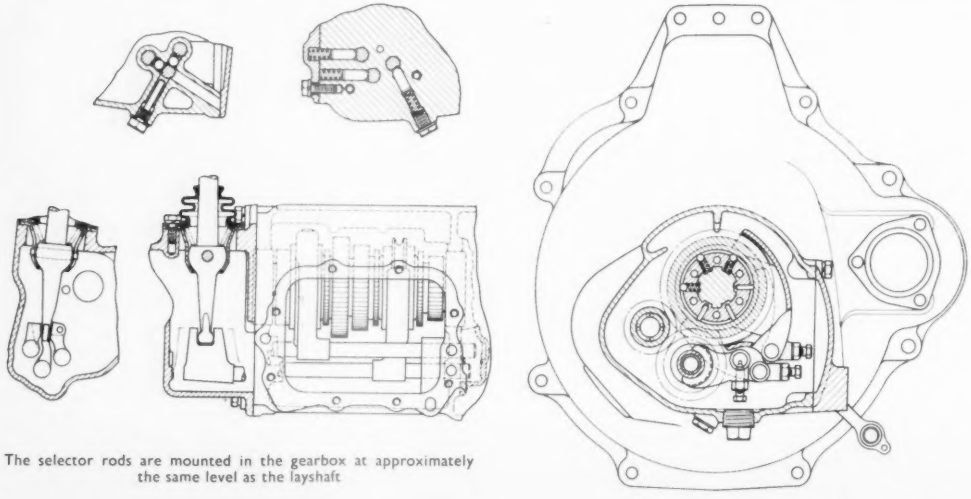
An interesting and ingenious feature of the arrangement of the Hobourn Easton oil pump is that its four rotors are so phased, relative to the cams, that they tend to damp out vibrations of the camshaft. The die cast aluminium pump body is spigoted into the rear of the crankcase and, together with its die cast aluminium cover, is secured by four $\frac{5}{16}$ in diameter set-bolts. The $\frac{1}{2}$ in diameter En 32 driving spindle is only 2 $\frac{5}{16}$ in long. It is slotted at its forward end to engage the peg diametrally positioned through the end of the camshaft. At the rear end, another $\frac{5}{16}$ in diameter peg of En 16 is pressed diametrally into the spindle and the rotor to supply the drive. Both the inlet and the outlet are well above the axes of the rotors so that the pump remains indefinitely primed with oil. A plug screwed into the crankcase side wall to the right of the pump, may be removed to allow oil to be inserted through a drilled passage communicating with the pump, for priming it after assembly.

An 18 s.w.g. pressed steel sump of six pints capacity is fitted. It is secured, together with a Langite joint washer, to the lower face of the sump by twelve $\frac{1}{4}$ in diameter set-bolts. There is a semi-circular cut-out to clear the main bearing caps at both the front and rear end. Sealing is effected by a rectangular section rubberized cork strip in a housing formed by spot welding a semi-circular portion of a dished ring to the outer face of the sump.

An inverted cupped pressing with gauze on the base as well as on the sides, forms the oil strainer. The base is removable. It is held in position by a central nut and bolt and a tubular distance piece. Riveted to the upper face of the strainer is a 16 s.w.g. pressed steel bracket carried by two set-bolts, one in the intermediate bearing cap and the other in the rear one. In front of the filter and midway between the two ends of the sump there is a transverse baffle.

Oil is drawn from the strainer through a $\frac{1}{2}$ in diameter pipe to a union screwed into a boss on the crankcase wall mid-way between Nos. 3 and 4 cylinders. Thence it passes through a vertical drilled hole to a horizontal one, and then into the pump. The outlet from the pump discharges into a transverse passage along which oil flows across to the right-hand side of the engine. An oil relief valve is screwed into the end of this passage. It lifts at a pressure of 60 lb/in² to discharge the surplus oil back into the sump. The oil gallery is drilled alongside the lower ends of the cylinders. Its axis is $\frac{33}{32}$ in above that of the transverse hole and 4½ in above the axis of the crankcase. The transverse hole intersects the passage between the gallery and the rear main journal so that oil may pass down to the bearing and up to the gallery. Midway between the ends of the crankcase, a duct is drilled from the right, into the main oil gallery, to feed an A.C. or Purolator by-pass filter, the axis of which is 2 $\frac{7}{16}$ in below that of the gallery. This filter is screwed into a boss on the crankcase wall and discharges into the sump through a vertical passage in the crankcase.

Oil is supplied to the main bearings



The selector rods are mounted in the gearbox at approximately the same level as the layshaft

through ducts from the gallery. Each big end bearing is served by two oil passages. One is drilled directly from the adjacent main journal, and is fitted with a restrictor jet at its delivery end, while the other is a branch drilled into it from a point on the periphery of the crankpin. The two passages are spaced approximately 130 deg apart. By means of this patented arrangement oil is fed to what, during the power stroke, is the pressure side of the bearing. It has the advantage that not only do the bearings last longer, but they also wear round instead of oval. Jets in the big ends of the connecting rods throw oil into the cylinder bores.

From an annular groove round each main journal bearing, oil passes upwards through another duct to the adjacent camshaft bearing. Through a slotted hole in the front camshaft bearing shell, oil is fed into an annular groove round the bearing. A flat machined on the bearing allows an intermittent supply of oil to pass from the groove into a vertical hole drilled up to the cylinder head. Thence the lubricant passes through a horizontal duct and a vertical duct to the front rocker pedestal, and then into the rocker shaft. Radial holes in the shaft supply the rocker bearings. The oil return to the sump is through the push rod housing. There is no jet tube to feed oil on to the timing chain, which is lubricated in the following manner. A second flat on the camshaft bearing communicates, once every revolution, with the forward end of the slotted hole in the bearing shell, and feeds oil to an annular space formed by the clearance hole in the front engine plate, into which the camshaft bearing projects. The space is closed at the front end by the camshaft thrust plate, through which a $\frac{1}{8}$ in hole is drilled to feed the oil on to the timing gear.

Ignition equipment

Lucas ignition equipment is fitted throughout except for the spark plugs, which are Champion NA 8 long-reach type. A standard 12 volt Q 12 coil supplies a DM 2 contact breaker and distributor unit having a centrifugal and vacuum controlled automatic advance and retard mechanism. The contact breaker and distributor unit is mounted in a flanged tubular adaptor, which is held by one countersunk setscrew, in a boss on the crankcase wall.

GEARBOX DATA

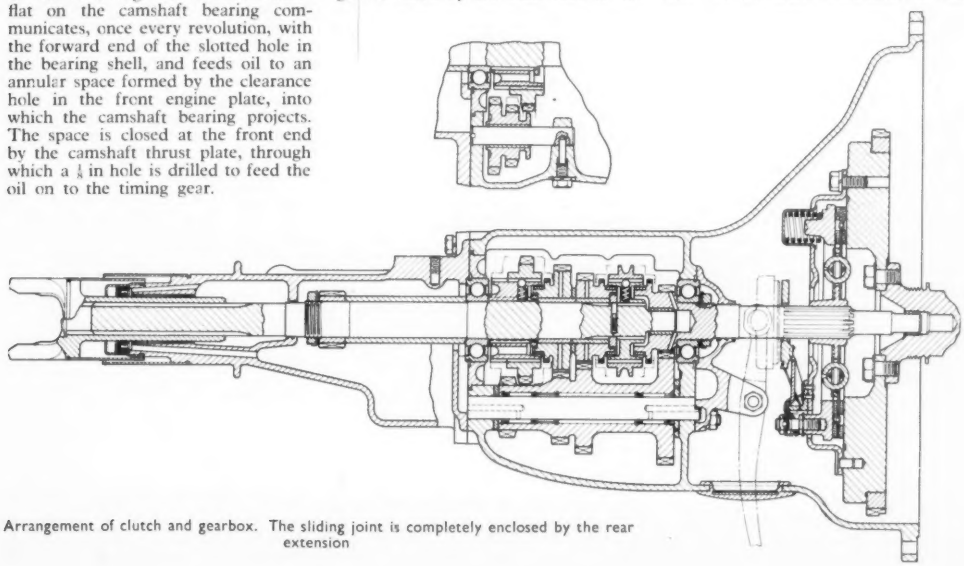
	Outside diameter	Helix angle	Diametral pitch
Primary gear	1·948 in–1·953 in	40 deg L.H.	13·054 normal
3rd speed gear	2·548 in–2·553 in	40 deg L.H.	13·054 normal
2nd speed gear	3·048 in–3·053 in	40 deg L.H.	13·054 normal
1st speed gear	3·465 in–3·470 in	—	10
Layshaft cluster :			
Layshaft gear	3·048 in–3·053 in	40 deg R.H.	13·054 normal
3rd speed gear	2·448 in–2·453 in	40 deg R.H.	13·054 normal
2nd speed gear	1·948 in–1·953 in	40 deg R.H.	13·054 normal
1st speed gear	1·625 in–1·630 in	—	10
Reverse idler	2·125 in–2·130 in and 1·695 in–1·700 in	—	10

The purpose of this adaptor is to retain the En 15 drive spindle, with its integral spiral gear, when the distributor unit is removed for servicing. This obviates re-timing the ignition when the distributor and contact breaker unit is replaced. A flange is formed round the slotted upper end of the spindle so that it cannot pass through the adaptor when the unit is removed. The other end of the spindle, machined to $\frac{1}{2}$ in diameter, is carried in a drilled boss in the crankcase wall on the other side of the engine. End thrust of the spiral gear is taken by the inner face of the boss against which it bears. A vertical hole drilled through the boss is provided for splash lubrication to the bearing surfaces.

Clutch and gearbox

A standard Borg and Beck 6½ in diameter, single dry plate clutch is employed. It has a sprung centre to absorb crankshaft vibrations and reduce shock loading on engagement. Operation is by means of a carbon bronze thrust ring. The unit is in an aluminium bellhousing cast integrally with the gearbox.

On the front wall of the gearbox, a die cast aluminium cover is secured by seven studs and nuts. This cover incorporates a forked extension to carry a $\frac{7}{16}$ in diameter hollow fulcrum pin for the clutch operating fork. The extension is offset to one side, therefore alternative covers must be provided for left or right-hand drive. A scroll type oil return is machined in the cover round the $\frac{27}{32}$ in diameter portion of the clutch shaft which is integral with the EN 34 primary gear. The primary shaft is carried in a ball race housed in the front wall of the gearbox and located by a snap-ring in a groove round its outer race. This snap-ring seats in a recess machined in the front wall. It is held in position by the cover. The inner race of the bearing is pulled up against the gear by a ring nut. A rearward extension of the



Arrangement of clutch and gearbox. The sliding joint is completely enclosed by the rear extension

primary gear carries the grooved phosphor bronze inner synchromesh cone. It is also machined to form the engagement teeth.

Immediately in front of the splines for the synchronizer, the main shaft is machined down to $\frac{9}{16}$ in diameter to fit in an Oilite bush pressed into the primary gear. The outer synchro-cones are integral with the case-hardened steel synchronizer hub. The included cone angle is 16 deg. Three spring loaded balls seat in a groove in the sleeve to provide the predetermined load for engaging the cones.

Aluminium bronze synchromesh cones are pressed on to the hubs of the second and third speed main shaft gears. The engagement teeth are machined integrally with these gears. These gears are carried on a phosphor bronze bush mounted on the main shaft. A phosphor bronze washer acts as a distance piece between the gears and between the two bushes. Two tongues in each bush engage in slots in the washer. The front end of the bush for the third gear is internally splined to engage the splineways in the main shaft. An En 34 thrust washer is carried in a groove in the shaft at the front end of the third speed bush. It is located endwise by the shaft splines and prevented from rotating by a peg and spring housed in a radial hole in the shaft.

As in the case of the other synchronizer unit, an aluminium bronze synchromesh cone is pressed on to the second speed gear boss, on which the engagement teeth are machined. Again, three spring loaded balls provide the necessary force to engage the cones. Integral with the sleeve member is a gear common to both first and reverse speeds. This gear is out of engagement when the sleeve is in the forward position for second speed; when removed to the rear it engages first speed. In the neutral position, the reverse idler may be moved forward to engage both this gear and the first speed pinion on the layshaft.

The rear of the main shaft is carried in a ball bearing housed immediately behind the splined collar, in an aluminium die casting spigoted into the rear wall of the gearbox. This separate housing is incorporated in order that the main shaft assembly may be inserted from the rear. The inner race of the bearing is clamped between a shoulder formed by splines on the main shaft and a distance sleeve $3\frac{5}{8}$ in long. Behind this is an En 32B gear for the speedometer drive, the whole assembly being secured by a ring nut locked by a tab washer, pulled up against the speedometer gear. A snap-ring in a groove round the outer race of the bearing, seats in a recess in the housing. Both the snap-ring and the housing are retained by a cast aluminium rear extension which is secured to the rear face of the gearbox by eight $\frac{1}{4}$ in diameter set-bolts, locked by spring washers.

An En 32B spindle $\frac{1}{2}$ in diameter × $2\frac{7}{8}$ in long, carries the En 24 reverse idler. It is located by a dowel-ended

A one-piece differential cage is employed in the final drive

set-bolt that registers in a diametral drilling in its front end, where it is supported in a boss in the wall of the gearbox. The other end is carried in the rear wall. Pressed into the idler gear is a Clevite bush $1\frac{1}{4}$ in long.

The En 34 layshaft cluster is on a $\frac{3}{4}$ in diameter En 36 spindle. This spindle is supported in two bosses, one in the front and the other in the rear wall of the gearbox. Axial movement rearwards is prevented by the rear extension casting. Part of the front end is machined away to leave only a semi-circular section. A rectangular boss cast inside the front cover projects into this cut-out so as to provide axial location, as well as to secure it against rotation. The layshaft cluster with its four integral gears is carried on two needle roller bearings $\frac{3}{4}$ in long mounted on the spindle. Each of the bearings, which are spaced $2\frac{13}{16}$ in apart, are located between circlips in the bore. Between the rear bearing and its front circlip, a distance tube is fitted so that the circlip groove may be positioned where the section is sturdy. The thrust is taken by two $\frac{1}{8}$ in thick En 24 washers, one against the front and one against the rear wall of the gearbox.

The oil capacity of the box is $2\frac{1}{2}$ pints. Two axial drilled holes, one from each end of the layshaft, connect with radial holes to form oil passages to the bearings. The axial holes are fed from two passages, one in the rear extension and the other in the front cover. Both convey oil to the layshaft bearings. Radially drilled holes in the primary gear, as well as in the third and second speed gears, pass oil to their respective bushes.

In the rear extension there is a transverse web immediately behind the ring nut securing the speedometer drive. It is bored to support the main shaft and has an oil return scroll machined in its periphery. To the rear of this web is a sliding propeller shaft joint, the sleeve member of which is carried in a Vandervell D2 bush. A pressed steel housing fitted on the end of the gearbox extension carries a felt dust excluder and a lit-type oil seal. Oil that is stopped by the seal is returned through a drilled duct to the chamber in front of the transverse web through which it passes back to the box. A second duct is incorporated to allow air to replace oil that drains away.

The complete sliding joint assembly is further screened against the entry of dust by two shroud sleeves. One, pressed on to the boss of the universal joint, extends forward inside the second sleeve, which is pressed on the gearbox rear extension. The front end of the outer sleeve and the rear end of the inner sleeve are each rolled into a groove to locate them positively against axial movement. A domed disc seals the rear end of the outer sliding member, with which is incorporated the front fork of the universal joint.

To keep down the height of the unit, the striker lever is offset approximately 1 in to the right of the centre of the longitudinal axis of the box. It is housed alongside the main shaft in the rear extension. The three En 8D selector rods are positioned on the same side approximately at the level of the layshaft. They are carried in bosses in the front and rear walls of the box. Access to the selector forks is through a cover bolted to the side of the gearbox casing

To form the gear locks, spring loaded balls housed in the front wall of the gearbox engage in grooves in the rods. In the rear wall, a single ball between each pair of rods, except where two balls are employed between those for reverse and first and second, engages in grooves in the rods to form the interlock mechanism. The striker forks are integral with the rear ends of the rods. In assembly, the die-cast aluminium bronze selector forks are threaded on as the rods are passed in through the rear wall. They are secured by cone-ended set-bolts fitted with lock nuts, the cones registering in tapered holes in the rods. Reverse is struck by lifting the gear shift lever so that the striker lever passes over the baulking shelf.

Back axle

The final drive ratio is 5·14 : 1. A Hardy Spicer propeller shaft 2 in diameter × $40\frac{1}{4}$ in long transmits the drive to the hypoid bevel, three-quarter floating rear axle, which is carried in a banjo-type casing. All gears and pinions are made of En 34. The crown wheel pinion, integral with its shaft, is overhung from two tapered roller bearings spaced about $1\frac{29}{32}$ in apart by a barrel shaped, cast distance tube. This is used in preference to a plain tube, as it is less expensive. Between the distance tube and the inner race of the front bearing are shims, by means of which the pre-load on the bearings is set. At the correct pre-load, the nip on the bearings is 0·002 in. The inner races of the bearings are clamped between the crown wheel pinion and the splined boss of the companion flange for the

universal joint, by a nut on the $\frac{1}{4}$ in diameter threaded front end of the shaft. A tab washer prevents the nut from working loose.

A Gaco oil seal round the boss is housed in a die-cast cover, which is fastened to the front of the cast aluminium nose piece. The seal housing is shrouded by a dished, pressed steel ring mounted on the companion flange boss. No provision is made for the adjustment of the assembly. It is located by the outer races of the bearings. The inner races are pulled against shoulders in their housing. The axis of the pinion is offset $\frac{1}{4}$ in to the right of the centre line of the car, and $1\frac{1}{8}$ in below the axis of the crown wheel.

The $6\frac{1}{2}$ in diameter crown wheel is secured to the one-piece malleable cast-iron differential cage by five $\frac{5}{16}$ in diameter set-bolts. A $\frac{9}{16}$ in diameter En 36 spindle for the two pinions is pressed into the cage. It is positively located by a $\frac{5}{32}$ in diameter peg in a hole drilled diametrally through one end of the spindle and through the cage. The end of the hole is peened. Two flats on the spindle feed oil from the centre to lubricate the bearing surfaces in the pinions, the spherical outer faces of which bear on to bronze thrust washers between the pinions and cage. On each side of the cage, bosses are machined $1\frac{1}{4}$ in diameter. They carry the ball thrust bearings supporting the cage in the nose piece casting. The outer races of these bearings locate against shoulders in their housing.

Shims are fitted between the shoulder and the race on the left-hand side for adjusting the mesh of the crown wheel, and on the right-hand side for adjusting the bearing pre-load. The pre-load is correct when there is a 0·002 in nip on the bearings. Each bearing cap is held by two $\frac{1}{4}$ in diameter studs, and projections on the outer face bear against the axle casing to prevent the cap from springing outwards under load.

Oil is fed to the bearing surfaces of the $2\frac{1}{2}$ in diameter differential gears through holes drilled in the differential cage. The gear bosses, which are $1\frac{5}{16}$ in diameter $\times \frac{3}{4}$ in long, bear directly in the cage, as do their flat thrust faces. Positioned $\frac{7}{8}$ in apart, the inner ends of the En 16 half shafts are splined into the differential gears in the usual manner. For a length of about 1 in they are supported in the differential cage where it is carried by the ball thrust bearings. Thus, bending loads in the shaft are reacted directly through the cage instead of through the splines and the differential gears to the cage.

A divided baffle ring of channel section is welded inside the banjo casing. Eight $\frac{1}{4}$ in diameter studs carry the nose piece casting. These studs are screwed into holes tapped through the axle tube, the baffle and a $\frac{1}{8}$ in thick reinforcing strip. Four of these strips are spot welded to the inner faces of the flanges of the baffle, each carrying two studs. The 16 s.w.g. pressed steel rear cover is welded on. A drain plug is fitted at the bottom of the banjo casing, while the combined oil level and filler plug is situated in the nose casting. S.A.E.90 oil is recommended. The capacity is $1\frac{1}{4}$ pints.

Two U-shaped brackets are welded underneath the tube, one to the right and one to the left, to suit right or left-hand drive requirements. The pivoted swinging link compensator for the hand brake mechanism is bolted to the appropriate bracket. Outboard of these are other U-brackets forming the spring seat. The U-bolts for pulling the spring up to the bracket, seat on channel section flanges at the base of another bracket. This bracket is welded on top of the tube and bolted to it is the bump stop rubber. The forward extension of the inner vertical wall of the spring seat bracket provides for the attachment of the rear shock absorber link.

The En 3A wheel hub carriers are butt welded to the outer ends of the axle tube. Five $\frac{5}{16}$ in diameter nuts and bolts hold each brake back plate, which is spigoted to a flange round the hub carrier. During road wheel removal the cast Conychrom iron brake drum is held to the headed flange at the end of the half shaft by one set-screw. This screw is carried in tapped holes in the housing for the wheel ball bearing. Another $\frac{1}{4}$ in diameter screw holds the half shaft to the wheel bearing housing during brake drum removal. Also screwed into the housing are the four $\frac{1}{4}$ in diameter set-bolts securing the wheel. These bolts have conical seatings under their heads. The inner race of the ball bearing is mounted on the $1\frac{1}{8}$ in diameter end of the hub carrier. It is held in position by a ring nut together with a large diameter lock washer. An oil return scroll working round the half shaft, is machined inside the hub carrier. Round the hub carrier is a Gaco oil seal, adjacent to the inner face of the ball bearing, carried in the bearing housing.

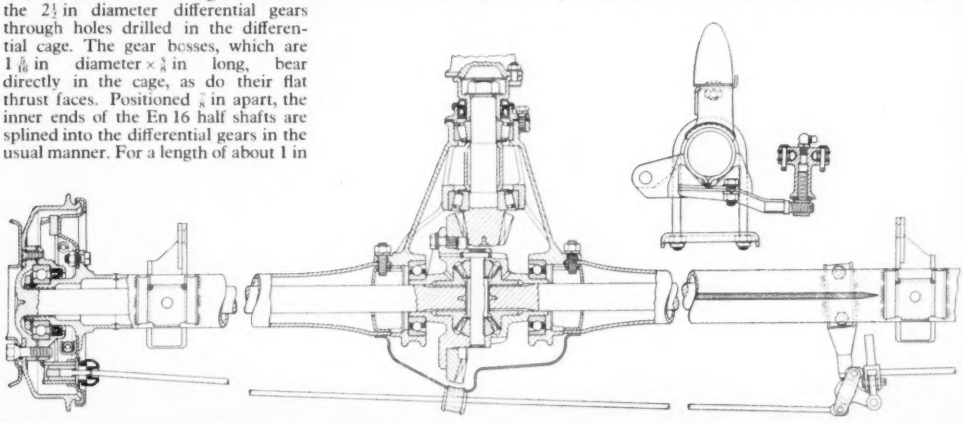
Rear suspension

Semi-elliptic springs 36 in long are employed at the rear. They are each made up from eight leaves $1\frac{1}{2}$ in wide and $\frac{3}{32}$ in thick. Between the top three, zinc inter-leaves are fitted. Metalastik rubber bushes are fitted in both spring eyes and bronze bushes in the shackle. Under full load, the rear end rate is 93 lb/in and the periodicity 82 cycles/min. The deflection to the fully laden position is $5\frac{1}{4}$ in, and to full bump it is $8\frac{1}{2}$ in.

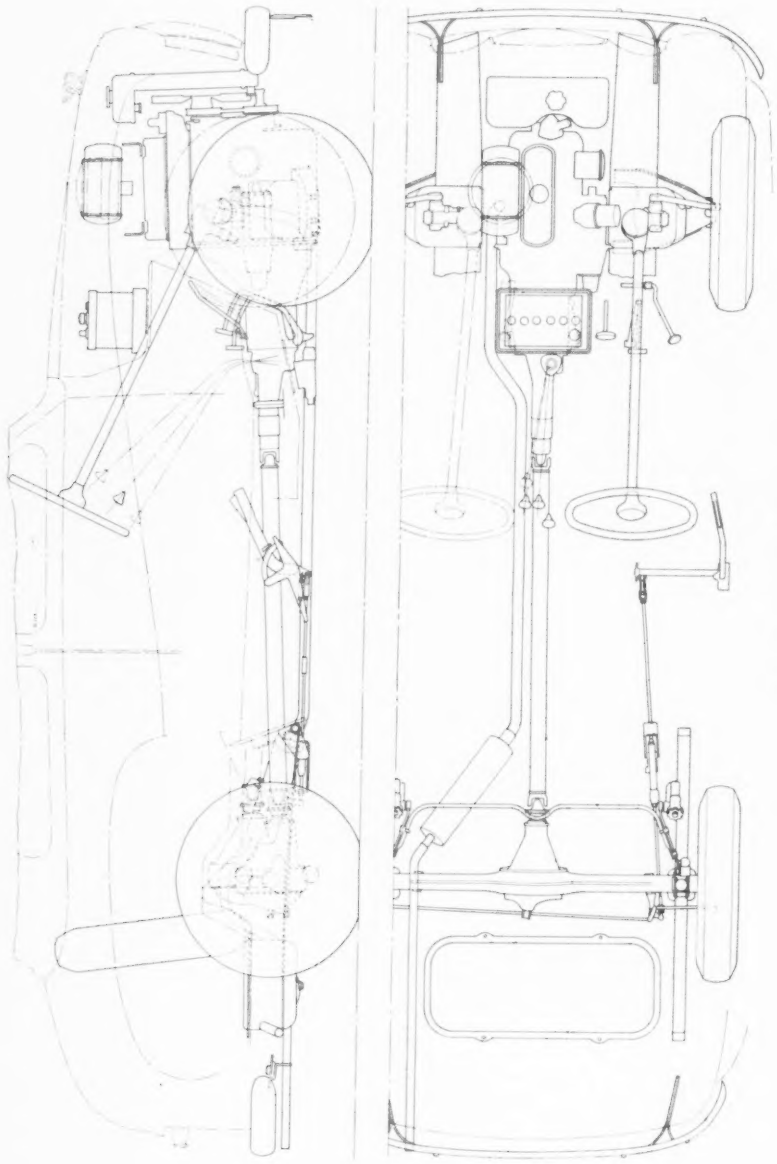
Armstrong double-acting hydraulic, lever-arm type shock absorbers are employed. They are mounted on the inner face of the wheel arch, which is suitably reinforced by a bracket. A $\frac{1}{2}$ in diameter anti-roll bar couples the two lever arms.

Front suspension

A coil spring and double transverse link type of front suspension is fitted. The front end rate is $67\frac{1}{2}$ lb/in and the periodicity 76 cycles/min. This is obtained with a spring having a rate of 271 lb/in. Deflection of the wheel is 6·06 in to the laden position, and 9·06 in to full bump. The effective length of the lower wishbone is $10\frac{7}{8}$ in, while that of the upper link, which is a single arm, is $7\frac{1}{4}$ in. Between the centres of the two lower link bearings the distance is $8\frac{7}{16}$ in. These bearings are $7\frac{3}{8}$ in below and $4\frac{11}{32}$ in inboard of the axis of the shock absorber spindle, which is the pivot

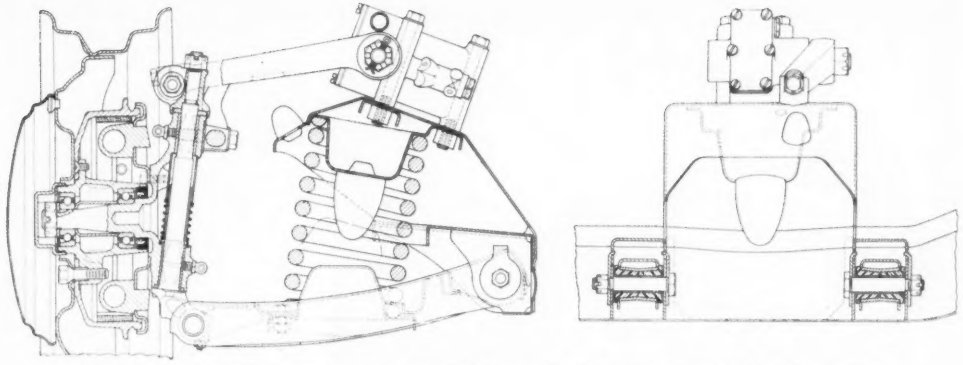


Three-quarter floating rear axle. The half-shafts bear directly in the differential cage



CHASSIS ARRANGEMENT

Wheelbase 6 ft 7½ in. Track: front 3 ft 9¼ in, rear 3 ft 8¾ in



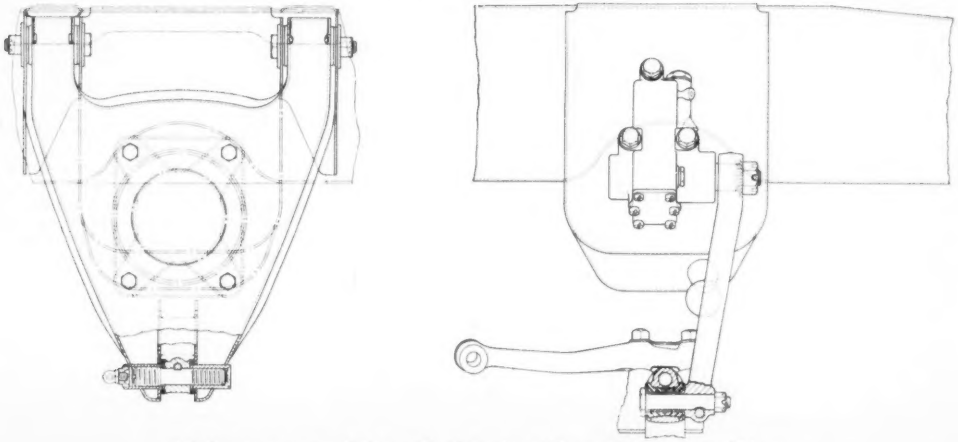
While maintaining a light construction, a wide spacing of the swivel pin bearings is obtained in the front suspension

for the upper link. The upper and lower bearings at the outer ends of these links are spaced 8¼ in apart. A castor angle of 2 deg (car empty) is incorporated, while the camber angle is 1 deg. The swivel pin angle is 6½ deg from vertical. On full lock the wheel angles are 31 deg and 24 deg, giving a turning circle of 35 ft. The toe-in is $\frac{1}{16}$ in.

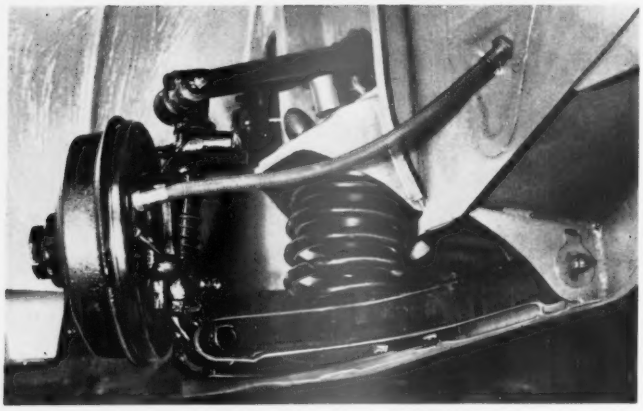
Armstrong double-acting lever-operated shock absorbers are employed. They are mounted above the spring on a 14 s.w.g. En 2 pan-shaped pressing. Reinforcement is provided beneath each pan, where the nuts for the three $\frac{3}{8}$ in diameter holding down bolts are welded on. This reinforcement takes the form of a channel section for the two outer bolts, while for the inner one the dished pressing round which the spring bears, is extended to provide the reinforcement. A rubber bump stop is mounted, coaxially with the spring, on the base of the dished pressing. With this arrangement the nearly vertical loads of the spring, bump stop and shock absorber have a common line of action. The rebound rubber is mounted on the outer lip of the 14 s.w.g. pan. In the full rebound position, it arrests the downward movement of the upper transverse link.

The cross section of the upper link is rectangular, $1\frac{3}{8}$ in wide by about $\frac{7}{8}$ in minimum depth. This link is splined on to the $\frac{7}{8}$ in diameter spindle of the shock absorber and secured by a split pinned slotted nut. The outer bearing of the link is trunnion mounted on a $\frac{1}{2}$ in diameter En 3 fulcrum pin. Clamped between the hexagon head on the rear end of the pin and the boss at the end of the transverse link is the trunnion boss, in which steel-sleeved rubber bushes are fitted. The fulcrum pin is secured in the upper link by a split pinned slotted nut on its $\frac{3}{8}$ in diameter front end. It is locked against rotation by a $\frac{5}{16}$ in diameter pinch bolt registering in a groove on the pin. The En 14B trunnion boss is pivoted on a $\frac{7}{16}$ in diameter upward extension of the swivel pin. A split pinned slotted nut pulls it down on to a shoulder immediately above the upper bearing where the pin is $\frac{3}{8}$ in diameter. Between the trunnion boss and the boss for the upper bearing is an Oilite thrust washer.

The lower wishbone is formed by a single 14 s.w.g. En 2A pressing. Beneath it is the dished pressing which seats the lower end of the spring, and on which the bump stop bears in the full bump position. It is therefore possible to withdraw the spring without removing the lower wishbone or any other part of the suspension. Metalastik rubber bushes are fitted to the inner wishbone bearings, which are carried in two channel section lugs welded to the front side members of the chassisless structure. These bushes are carried in a tube welded between the downward turned flanges of the wishbone pressing which, at each of these two points, is a channel section. The base of the channel is wrapped round between these two flanges and welded to them, partly encircling the bearing to give stiffness in the longitudinal direction. Correct compression of the rubber bush is applied by a $\frac{1}{2}$ in diameter En 14B shouldered bolt and nut. The shoulder bears against the outer flange of the lug welded to the front side member, while the head of the bolt



A single transverse upper link is employed together with a pressed steel lower wishbone



The front suspension unit is accommodated in a vertical channel in the wheel arch

bears against the inner one. Interposed respectively between each lug and the head of the bolt and nut is a plain washer. To prevent rotation of the pins, each of these washers has a tab which is bent so as to engage in a rectangular hole punched in the appropriate lug.

A screw type outer wishbone bearing is employed. Its thread is cut in a two-piece $\frac{3}{4}$ in outside diameter tube. The outer end of each piece is carried in the downwardly turned flanges of the wishbone pressing, while the inner ends are supported by a separate channel section bracket welded on. Carried between the two inner ends is a boss forged integrally on the lower end of the swivel pin. Mounted in this boss is the En 33 fulcrum pin, held by a $\frac{1}{4}$ in diameter locating cotter. In order that the pin may be screwed in from the rear, its front end is threaded $\frac{9}{16}$ in diameter, while its rear end is $\frac{11}{16}$ in diameter. The front end of the bearing tube is sealed by a plug pressed and peened in, while screwed into the rear end is an adaptor for a grease nipple. Round the inner end of each tube between the support bracket and the swivel pin boss is a felt ring.

Two Clevite bushes $1\frac{1}{2}$ in long by $\frac{11}{16}$ in inside diameter support the En 33 swivel pin. Their centres are $4\frac{1}{2}$ in apart and they are carried in bosses formed integrally with the En 14B stub axle. Between these two bosses, the swivel pin is shrouded by a telescopic tube, the two parts of which are held, one against each boss, by a compression spring. This spring is positioned round the inner tube and seats on a pressed-out flange on its lower end. The other end of the spring bears against a similar flange on the lower end of the outer tube. This layout is better than the older and more conventional knuckle joint arrangement in that it permits the swivel pin bearings to be more widely spaced, while keeping the unsprung weight to a minimum.

Each end of the bosses for the swivel pin bearings is drilled and tapped for a grease nipple. The grease is passed through a hole in the bush into an annular groove round its inner periphery and is spread over the bearing surface by star grooves. A cork sealing ring is fitted in a groove round the bottom face of the lower boss; it bears against the shoulder formed on the swivel pin by the wishbone bearing boss.

The En 3A wheel hub is carried on two combined ball journal and thrust bearings, the inner races of which are spaced $1\frac{1}{2}$ in apart by a distance piece of 14 s.w.g. pressed steel, thimble shaped to give it a certain amount of resilience. A nut on the outer end of the axle pulls up the hub assembly. Interposed between this nut and the inner race of the outer bearing is a plain washer. On its inner periphery, it has a tongue which engages in a slot machined in the portion of the stub axle on which the front bearing is carried. The pre-load, which is correct when the nip on the bearing is 0·002 in, is checked by measuring the torque required to turn the wheel. The outer races of the bearings are carried in housings machined, one in each end of the hub. At the inner end, the housing also carries a Gaco oil seal bearing on the $1\frac{3}{8}$ in diameter portion at the root end of the stub axle. Immediately outboard of this, the axle is machined down to 1 in diameter, forming a shoulder against which the inner race bears.

The brake master cylinder and the pedals are mounted under the side member

Grease is packed into the hub on assembly and is retained by a pressed or die-cast aluminium cap screwed on to a thread machined round the outer end of the hub. An effective seal is ensured by fitting a Vellumoid washer inside the cap. The brake back-plate is spigoted on to the flange round the stub axle and held by four $\frac{1}{4}$ in diameter set-bolts, locked by spring washers.

Steering

The steering gear, supplied by Cam Gears Ltd., has a ratio of 12:1. Two bosses are formed on the steering arm, one for each of the two $\frac{5}{8}$ in diameter set-bolts that secure the arm to the upper swivel pin bearing boss. The effective length of the arm is about $6\frac{5}{16}$ in. The side rods are $\frac{1}{2}$ in diameter $\times 7\frac{5}{8}$ in long and have a ball joint at each end. At the outer end, the ball joint is mounted on the steering arm, while at the inner end it is carried, on one side by the drop arm and on the other by the idler lever. Both the drop arm and the idler lever are in the form of a bell crank with the arms at an angle of about 14 deg. The second arm of the bell crank lever carries the ball-jointed end of the adjustable track rod tube, which is $\frac{3}{4}$ in outside diameter. Lock nuts on the end fittings secure them after adjustment has been made.

Pressed steel brackets of identical design, positioned one on each side of the vehicle, carry the steering box and the idler lever. A tubular extension of the steering box, housing the spindle for the drop arm, is carried in a split boss secured to the bracket by three $\frac{3}{8}$ in bolts screwed into a $\frac{1}{4}$ in thick tapping plate. One of the three bolts registers in a groove on the tubular extension and clamps the ends of the divided boss to hold the steering box firmly in position. The cylindrical idler body is handed for left and right hand drive vehicles, but is carried in a similar boss to that in which the steering box is mounted. The steering box is not handed.

Brakes

At the front, Lockheed hydraulic brakes with two leading shoes are fitted, while at the rear, Lockheed mechanical units with two trailing shoes are employed. The total shoe area is 61 in^2 and the shoe is $1\frac{1}{4}$ in wide. Cast Conychrom brake drums 7 in diameter are employed on all brakes. The rear brakes are mech-

anically actuated through a pivoted swinging link type compensator, but the main operating cable extends forward to a small hydraulic actuating cylinder mounted on a bracket on the heel board. This cylinder is operated by a control rod which has on its front end an adjustable slotted fork end connected to the hand brake lever.

The hand brake control is a simple pull-up lever with a trigger-type ratchet. It is mounted immediately outboard of the driver's seat on to a cross tube. Welded to the other end of this cross tube is the lever for operating the control rod, which is adjacent to the propeller shaft tunnel. The ends of the tube are carried in graphite impregnated spherical bearings. Each of these bearings is clamped between a small cover plate, bolted at the outer end to the sill plate and at the inner end to a bracket bolted under the foot ramp beneath the front seat. As compared with purely hydraulic operation, this arrangement avoids the use of an appreciable length of metal piping and two flexible pipes. Moreover, there are no hydraulic cylinders inside the brake drums. At the same time, the desirable feature of differential front and rear braking has been incorporated. Further economy has been effected by upsetting the eyes on the ends of the compensator rods, instead of using screwed fork ends.

The brake pedal is mounted on a spindle supported in a combined bracket and master cylinder bolted to the foot ramp. This bracket also forms the reinforcement to which the master cylinder is bolted. A lug, projecting downwards from the boss at the lower end of the pedal, has two holes drilled in it. The upper one carries the fork end of the master cylinder plunger. A grease nipple is screwed into the end of the brake pedal spindle, the lubricant being fed through drilled holes to the phosphor bronze bush pressed into the pedal boss. The clutch control pedal is mounted on a spindle supported at its outer end in a spherical bearing similar to those used for the hand brake cross tube. In this case, however, the bearing is mounted in the vertical flange turned downwards on the side member. The other end of the spindle is supported on the gearbox casing. A lever is welded on or near the inner end of this spindle. It operates, through an adjustable rod, a clutch withdrawal lever carrying the carbon clutch throw-out bearing.

Lucas 12 volt equipment is used throughout. A G.L.T.W.7A battery of 32 amp-hr capacity and a 10 hr rate is employed. It is served by a C 39 PV/2 dynamo operating in conjunction with an RB106 cut-out and regulator. The system is protected by an SF6 fuse unit. Other electrical equipment includes an M35G starter motor, a single 1235J horn and SF34 direction indicators.

THE PREVENTION OF SCUFFING

IT is known that scuffing can be prevented by giving careful attention to machining, assembling and initial running-in. One essential requirement is, however, that the mating surfaces must be adequately lubricated from the first moment that they are in motion until a condition of surface is obtained that is proof against scuffing.

The lubricating properties of graphite are well-known, but in recent years a close evaluation has been made of the extent of these lubricating characteristics. It can be shown that the best results are obtained if the graphite is very finely divided so that when the particles lie on the metal faces they are orientated with their slip planes parallel to the direction of the rubbing. Extreme fineness of the particles also enables the graphite to adhere to, or to be absorbed by, the metal surface and to form a thin lubricating film which is referred to as a "graphoid surface."

Numerous measurements of the coefficient of friction of orientated films of pure graphite show that this lies in the vicinity of 0·1. This value is for a graphited surface free from oil or any other liquid. It represents a good measure of lubrication and enables dry films formed from dispersions of colloidal graphite in volatile media to be used in industry for dry lubrication where oil and grease may be undesirable. When oil is present on a graphoid surface, the coefficient of friction lies around 0·05 under boundary conditions of lubrication. Under similar conditions the value for oil without any graphite present on the surface would be higher. From the analysis of data obtained from various sources, it is shown that, in the presence of oil, the coefficient of friction obtained for two rubbing surfaces which have been treated with graphite is characteristically lower than that for the same rubbing surfaces without any graphite. This accords with the generally accepted view that much of the friction between rubbing faces is contributed by stick slip or the adhesion of metal high spots.

Possessing a melting point which, for all practical purposes, exceeds that of any metal, graphite will provide parting between the high spots on rubbing faces when normally they might fuse together. It is this ability of the microscopic particles of colloidal graphite to separate the high spots which possibly accounts for the lower friction obtained with a graphite surface in the presence of an oil film, to which reference has already been made. It is also possible that the better oil-spread yielded by a graphited surface as compared with plain metal will assist the reformation of the oil film after it has been broken by excessive load at particular spots on the bearing surface. In addition, it is known that a graphited surface has oleophilic qualities, which means that oil will adhere better to graphite by adsorption, in addition to yielding better spread.

It is these underlying advantages of a graphited surface that have led to the development and widespread use of dispersions of graphite to provide lubrication in various types of mechanical equipment. A method of treating new surfaces which can be said to render them more resistant to scuffing, is to apply to them a dispersion of colloidal graphite in a volatile carrier which, on drying, produces a hard, oil-resisting graphite film. This film prevents direct metal-to-metal contact between the adjacent surfaces, and is only gradually removed during the running-in period.

Tests carried out on internal combustion engine pistons which had previously been treated with this dispersion of colloidal graphite, show that without previous running-in, they can be subjected to several hours running under maximum load and speed conditions without tendency to scuffing occurring. The performance of graphite-coated pistons under these severe running conditions has convinced some well-known aircraft and other internal combustion engine manufacturers of the value of this treatment which has now been adopted as standard practice in their works.

The method by which pistons and other engine parts are coated is extremely simple. The appropriate dispersion of colloidal graphite, after dilution with a quantity of volatile solvent, is sprayed on to the degreased surface to be coated. The coating is dried in air for several hours or at a temperature of 100 deg C for one hour, and when cooled to room temperature the treated parts may be handled without danger of removal of the graphite coating. The thickness of the graphite film is of the order of 0·0001 in and does not, therefore, modify the dimensions of the finished parts to an extent which would necessitate any alteration in the design clearances.

For the treatment of other parts such as engine camshafts, crankshafts, sleeves and valve gear components, and laminated leaf springs, the dip method of application may be found to be more convenient. In all cases it is important that the components to be treated are thoroughly degreased and it is preferable to apply the colloidal graphite whilst the parts are hot enough to facilitate evaporation of the carrier liquid. Air drying or low-temperature baking will render the coating completely hard and oil-resistant.

Those interested in this process may obtained further details concerning the lubricating effect of colloidal graphite from the Journal of the Institute of Petroleum, Vol. 37, No. 329, May 1951, and from Acheson Colloids Limited, who are manufacturers of graphite dispersions.

II

CENTRIFUGAL FILTRATION

An Interesting Development by Glacier Metal Co. Ltd.

REMOVAL of contaminants from lubricating oil by centrifuging is not new; for example, centrifugal filters with either independent mechanical or electrical drives have been widely used for filtration on aircraft engines, but we believe that the centrifugal filter made by Glacier Metal Co. Ltd., Alperton, Wembley, Middlesex, is the first to be developed for use on internal combustion engines for road vehicles.

An outstanding feature of this filter is that the centrifuging action is dependent only upon the pressure in the lubricating system and it does not in any way entail an extra drive. Essentially, the filter is assembled from the parts shown in Fig. 1. The outer bowl A is connected to the oil supply line. Mounted in this bowl is a hollow spindle, which is in connection with the oil supply passage. The rotor B, is mounted on the hollow spindle. It has two stand pipes, each connected to a fine nozzle on the underside of the rotor.

Oil from the pressure supply line goes up through the hollow spindle into the rotor. As the oil accumulates in the rotor, it passes into the stand pipes down to the nozzles, through which it is expelled into the outer bowl. The rotor is caused to rotate by the reaction to the force set up by the oil as it is expelled from the nozzles as fine spray. With oil pressure at 30 lb/in^2, which is the recommended minimum, the rotor revolves at approximately 4,000 r.p.m. The centrifuged oil passes back to the sump through a central hole in the base of the outer bowl. The caps for the rotor and the outer bowl are shown at C and D respectively in Fig. 1.

Fig. 1. Component parts of the Glacier centrifugal filter

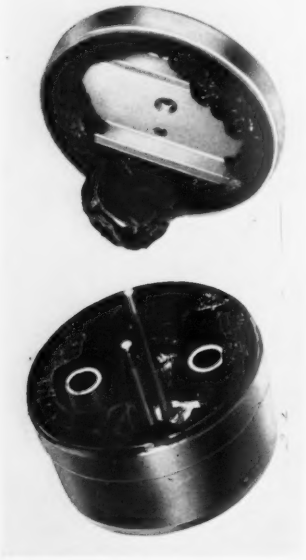


Fig. 2. Build-up of contaminants after 100 hours running with dirty oil

The centrifugal action causes the contaminant particles in the oil to be thrown against the sides of the bowl. At 4,000 r.p.m., the centrifugal force is great enough to deposit the particles on the sides of the rotor with sufficient adhesion to resist being washed off. Fig. 2 shows the manner in which the contaminants build up in the rotor. This deposit can easily be stripped off when the filter is to be cleaned.

In comparison with the types of filter more commonly used for road vehicle engines, many advantages are claimed for the Glacier centrifugal filter. They may be summarized as:—

(1) There is no element to be replaced.
(2) The filter is easily cleaned, neither special tools nor mechanical skill being needed.
(3) The filtering capacity is many times greater than that of existing pack-type filters.
(4) The filter cannot become clogged, even if it is used until the deposit is excessive.
(5) Filtration efficiency remains uniformly high and does not diminish appreciably as the dirt accumulates.
(6) The oil that is being filtered does not pass through previously collected dirt as is the case with pack-type filters.
(7) Inspection is easy. It is only necessary to remove the top cover and the rotor cover.
(8) It is not necessary to disturb oil pipes or oil pressure joints when the filter is to be cleaned.
(9) The filter can be mounted in almost any accessible position at the side or on top of the engine. Practically the only mounting limitation is that the filter must be within 20 deg of the vertical.

At present only one size of filter is available. It is intended mainly for diesel engines in the range from 60 to 120 b.h.p. It has a dirt capacity of 20 cu in, and is essentially a by-pass type. The flow rate is, however, such that the oil sump contents are filtered every few minutes.

COMBATING DAZZLE

ACCORDING to F. Verney in *Vie Auto*, May, 1952, with the usual headlamps a sharply defined light zone is produced above which the intensity of illumination is largely but not altogether eliminated. The dazzling effect is approximately proportional to the distance between the eye and the boundary between the zones, and is consequently stronger on the driver of a low passenger car. Momentary dazzling will also be produced through the pitching of the car, unevenness of the road, and changes in the slope.

It is suggested that the situation could be considerably improved by adopting the simple expedient of placing the anti-dazzle shield at the side of the filament instead of beneath it. In this way a car travelling along a line parallel to the axis of the road will cast a beam of light of full intensity along one side of the road without affecting the other side. With two cars approaching one another in the middle of their respective halves of an 8 m wide road, a dark zone more than 2 m wide remains in the middle of the road, protecting both drivers from the glare. A slight inclination of the shield from the vertical might be advisable to extend the illuminated band over the whole width of the car.

The dazzling of the driver is still possible on a curve or when the car temporarily abandons the position parallel to the road, as when overtaking. The system has been tested and has given very encouraging results. (*M.I.R.A. Abstract No.* 5892.)

ENGINE LUBRICATING OILS

The Performance of Premium and Heavy-duty Qualities

By J. G. Withers, B.Sc., A.M.I.Mech.E.

LUBRICATING oils for internal combustion engines have two primary functions to fulfil: the efficient lubrication of all mating surfaces, and the removal of heat from bearings, pistons, and other parts. In fulfilling these functions they are called upon to operate over a very wide range of conditions, and particularly so in automotive service. The operating temperature, which in a variety of ways affects lubricating oil performance, even excluding low temperature starting, varies widely from short-journey operation in severe winter conditions on the one extreme to heavy continuous duty at high temperature on the other extreme. In addition the oil may suffer contamination from condensed water, unburnt fuel, and carbonaceous partially-oxidized products from the combustion gases, apart from any changes that may result from its own shortcomings.

"Premium" quality oils are generally considered to be those which have definite advantages over the general run of "Regular" oils. Such advantages could include improved startability; reduced friction; improved oiliness; lower consumption; improved oxidation stability; improved engine cleanliness; and improved wear characteristics. These advantages accrue from the selection by the lubricating oil producer of a crude oil and refining process that give a product with the desired characteristics, supplemented by the judicious use of additives to impart special properties. Premium quality oils are recommended for use in all gasoline engines and for moderate duty in diesel engines.

"Heavy-duty" oils set an even higher standard than the Premium quality with a particular accent on engine cleanliness under the more arduous conditions of diesel engine operation; they are recommended where experience has shown that lower qualities are not wholly satisfactory.

It is the object of this article to discuss the aspects by which high-quality lubricating oils are judged—from the points of view of how they can be measured and the levels of performance that should be aimed at. In regard to the latter there can, of course, be no finality: the policy to be adopted should be one of continuous improvement towards higher and higher standards; but the availability of good lubricating oil should not be made an excuse for poor engine design or inferior maintenance.

Ease of starting

Ease of starting of an engine depends on many factors including: atmospheric temperature; size and condition of the starter battery; design, adjustment, and age of the engine; fuel quality; and lubricating oil quality. For any vehicle with a given fuel and lubricating oil there is a temperature below which it is impossible to start the engine: the mixture strength may be too weak because of carburettor limitations or inadequate fuel quality; the ignition system may not produce a sufficiently intense spark; or the engine cannot be turned at a sufficiently high speed.

The lubricating oil has, of course, a large influence on the breakaway torque and resistance to turning. Viscosity at the starting temperature is the major controlling factor, but the amount and form of wax crystallization is also relevant. An indication of starting quality is given by extrapolated viscosities at 0 deg F for the three winter grades—5W, 10W and 20W—in the S.A.E. Viscosity Classification[1]. Extrapolated viscosities are reasonable estimates of true viscosities and, therefore, of relative ease of starting if the pour point of the oil is not approached too closely, that is, provided that there is little wax

Fig. 1. Copper-lead bearing corrosion in a high-speed diesel engine. Heavy and dangerous corrosion of big-end and main bearing shells caused by the development of acids in the oil at high temperature

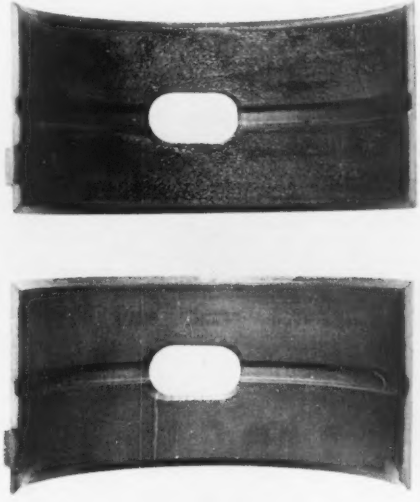


Fig. 2. Chevrolet L-4 copper-lead bearing corrosion
Upper : After a test on a Regular Oil
Lower : After a test on a high quality oil with oxidation inhibitor

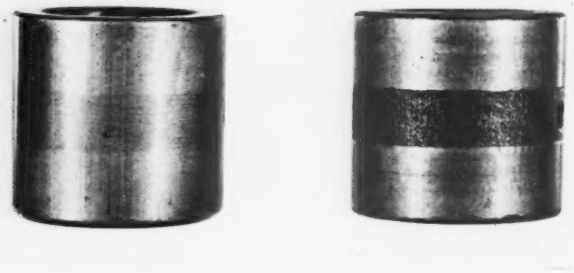


Fig. 3. Corrosion of phosphor bronze bushes
Left : Satisfactory combination between bearing material and oil additive for operation at high temperature (200 deg C)
Right : Corrosion of bearing by an unsuitable combination between bearing material and oil additive

separation. As a safeguard the pour point should be specified at least 15 deg F below the lowest temperature at which it is expected to use the oil. In the United Kingdom, therefore, pour points of 0 deg F should be quite adequate for all the winter grades.

Table I lists the S.A.E. extrapolated viscosity limits, together with estimates of the lowest engine temperatures at which reliable starting can be obtained in an average car for oils which only just meet the S.A.E. requirements, and pour points appropriate to starting at these temperatures.

Limits for the ratio of the viscosity at 100 deg F to the viscosity at 210 deg F are also given in Table I. As shown previously by the author[2] these are very convenient for ensuring that the extrapolated viscosities at 0 deg F shall not be exceeded.

The maximum viscosity limits at 0 deg F given in the S.A.E. classification for the 10W and 20W grades are considered to be too high for Premium and Heavy-duty oils; preferred limits have been included in Table I which, in the author's opinion, represent a minimum standard where cold starting is important. Good low temperature viscosity characteristics are desirable in a lubricating oil even though the lowest possible starting temperature is never reached; under all conditions lower viscosity means quicker starting and less wear and tear of the battery.

Running viscosity—Internal friction and oil consumption

The viscosity of the lubricating oil at the running temperature has a pronounced effect on the mechanical losses in the engine and, consequently on the overall efficiency. Roughly speaking, the losses increase as the fourth root of the absolute viscosity at 210 deg F. On this basis the engine friction losses will decrease by nine per cent for each 10 S.A.E. numbers reduction in viscosity grade. An improvement in fuel consumption of nine per cent will therefore be obtained at no load, but as the throttle is opened the importance of the friction losses becomes less and the gain in fuel consumption becomes correspondingly smaller. At one-eighth throttle, which is the lowest likely road condition, the gain is six per cent and at full throttle it may be as little as one per cent.

Thus, depending on the load factor —that is, the size of the engine in relation to the size of the vehicle and the type of driving—a reduced fuel consumption of between one and six per cent can be expected from a reduction in engine lubricating oil viscosity of 10 S.A.E. numbers; an average improvement is four per cent with an average of two per cent for the change from S.A.E. 10W to 5W.

Another advantage of using thinner lubricating oil is the increased rate of circulation which, together with the reduced friction, results in reduced bearing temperatures. The increase in circulation is roughly inversely proportional to the square root of the viscosity at 210 deg F provided, of course, that the oil pump has a sufficient reserve of capacity. In an actual engine test it was found that a change from S.A.E. 50 to 10W brought about a reduction in the big-end bearing temperature of 20 deg F for the same running conditions[3].

Better heat transfer coefficients and reduced piston deposits are also associated with oils of lower viscosity. Thus, in addition to easy starting, there are appreciable gains to be had from using the thinnest possible lubricating oil. What then imposes the lower limit in running viscosity? First, the provision of adequate lubricating value to avoid piston scuffing and bearing failure under heavy duty; and secondly, the increase in oil consumption associated with lower viscosity.

While there is a possibility that scuffing might occur with low-viscosity oils in the Regular class, Premium oils usually contain additives which impart additional oiliness so that in most modern engines even S.A.E. 5W may be used under summer conditions without scuffing. Reduced viscosity does, however, lessen the load-spreading action of the oil film in a bearing and increases the surface fatigue stresses. On the other hand, the reduced temperature of the bearing shell associated with the lower viscosity means that the bearing metal is better able to withstand the stresses. The author believes that S.A.E. 5W will not give bearing trouble provided that the oil pump has adequate capacity and provided that the sump-oil temperature is not excessive.

Oil consumption undoubtedly tends to increase as the oil viscosity at the working temperature of the piston rings decreases, but the relationship is not

TABLE I. STARTING VISCOSITIES OF MOTOR OILS

S.A.E. Viscosity No.	Maximum extrapolated viscosity at 0 deg F	Extrapolated viscosity at 0 deg F is satisfied if cs at 100 deg F / cs at 210 deg F is not more than	Lowest reliable starting temperature	Maximum pour point appropriate to lowest starting temperature
	centistokes		deg F	deg F
5W	870	5.40	−10	−25
10W	2,600 (2,000)	6·85 (6·50)	0 (−2½)	−15 (−20)
20W	10,500 (7,000)	9·25 (8·50)	+15 (+10)	0 (−5)

Note.—The figures in brackets are preferred values for Premium grades. The low pour points, however, are required only if the climate demands it.

TABLE II. RECOMMENDED VISCOSITY GRADES OF MOTOR OILS

	Winter-max. temperature 68 deg F (20 deg C)	Summer-max. temperature 104 deg F (40 deg C)	Tropical-max. temperature above 104 deg F (40 deg C)
New engines	5W	10W	20 or 20W
Engines in average condition	10W	20 or 20W	30
Old and worn engines	20W	30	40

a simple one. The evidence[4,5,6] seems to show that oil consumption is independent of viscosity until a critical value is reached below which the consumption starts to increase rapidly. In new engines this critical value is outside the working range so that all grades 5W to 50 give substantially the same rate of consumption. As the condition of the engine deteriorates the critical viscosity moves upwards until in the extreme there is a continuous decrease in consumption from 5W to 50.

As a general guide for engines in average condition it may be taken that S.A.E. 30, 40 and 50 give the same consumption; S.A.E. 20W, ten per cent increased consumption; S.A.E. 10W, forty per cent increased consumption; and S.A.E. 5W, eighty per cent increased consumption.

The running temperature of the cylinder wall varies in different engines between 260 and 400 deg F at the top, and 200 and 300 deg F at the bottom. Thus, oil consumption will tend to be less with oils of higher Viscosity Index, since the control temperature of 210 deg F is lower than the average working temperature. Premium and Heavy-duty qualities should have a V.I. not below 80. From the foregoing it is clear that the oil to be used is the lowest viscosity grade that does not give an excessive oil consumption; Table II gives general recommendations.

Table III gives the viscosity limits in the S.A.E. classification together with recommended mean viscosities based on the author's logarithmic system[2]. As a link with past practice, Redwood seconds at 140 deg F based on 85 V.I. are included, but it is recommended that this system is abandoned in favour of centistokes at 210 deg F for control and specification purposes.

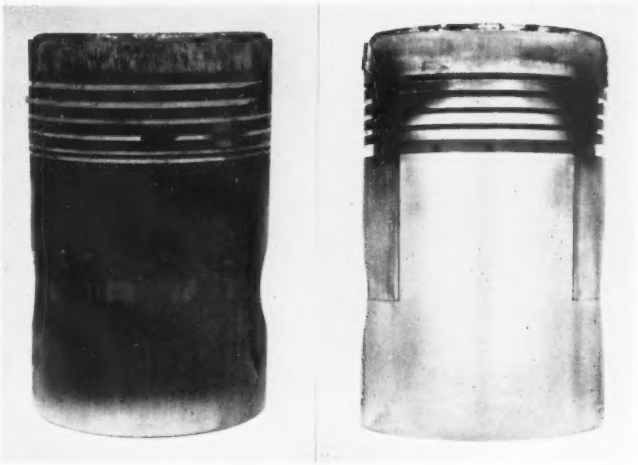


Fig. 4. High speed diesel engine pistons—thrust sides
Left : After a test on a Regular oil. Note heavy lacquering and stuck piston rings
Right : After a test on a Heavy-duty oil. The piston rings were free after test and have been removed

TABLE III. RUNNING VISCOSITY OF MOTOR OILS

S.A.E. Viscosity No.	Viscosity—cs at 210 deg F		Redwood secs at 140 deg F. for proposed mean viscosity at 85 V.I.
	S.A.E. limits	Proposed mean viscosity	
5W	Not less than 3·88	4·20	55
10W	Not less than 4·20	5·00	65
20W	Not less than 5·75	7·07	95
20	5·75— 9·65	7·07	95
30	9·65—12·98	10·00	145
40	12·98—16·82	14·14	220
50	16·82—22·75	20·00	350

Oxidation stability and oil deterioration

Lubricating oil in an engine can deteriorate for any of the following reasons:

1 Contamination by carbonaceous partial-combustion products from the fuel
2 Contamination by unburnt fuel
3 Contamination by lead compounds from lead tetraethyl in the fuel
4 Contamination by adventitious solid matter
5 Contamination by carbonaceous material formed in the lubricating oil
6 Contamination by lubricating oil additive ash
7 The development of corrosive properties to copper-lead bearings
8 Increase in viscosity due to oxidation.

Only the last four items are the direct responsibility of the lubricating oil, although additives, as well as dealing with carbonaceous material formed in the oil itself, can equally well deal with similar products from the fuel. Lead salts, additive ash, adventitious matter and some of the carbonaceous deposits from the fuel and lubricating oil are normally removed by the engine oil filter. Rapid choking, however, cannot be tolerated, and is usually a sign of inferior lubricating oil, bad engine condition, or too long an oil change period.

The use of a leaded fuel usually results in a cleaner engine, probably because of a mild polishing action on the rubbing surfaces. For the same reason copper-lead bearings are more susceptible to corrosion when the engine is run on a leaded fuel, as protective films may be rubbed away. Sludges, however, may be increased in magnitude by the use of a leaded fuel. Unburnt fuel dilution does not reach significant proportions in modern engines, and neither gasoline nor diesel fuel dilution in moderate quantities is believed to have any pronounced effect on engine behaviour, although partly burnt products are harmful.

When the temperature of the oil in circulation is high, which is likely when the engine speeds and ambient temperatures are high and when there is inadequate cooling, acids may be produced in the oil which attack copper-lead bearings. Such attack can reach the serious proportions shown in Fig. 1, and result in a major failure in the engine. The development of compounds in the oil which are corrosive to copper-lead bearings, and the increase in viscosity in use, can be examined by an engine test in the General Motors Chevrolet engine (C.R.C. designation L-4). This test was originally devised for testing Heavy-duty oils to the U.S. 2-104B specification, now superseded by the MIL-O-2104 specification. The author considers, however, that Premium quality oils should, in some markets, also be called upon to satisfy this test.

The L-4 test in the Chevrolet engine is run for 36 hours under the following conditions: speed 3,150 r.p.m.; b.m.e.p. 35 lb/in^2; lubricating oil temperature 280 deg F (265 deg F for S.A.E. 10W and low viscosity oils); and jacket temperature 200 deg F. These conditions are particularly severe with regard to the conditions in the crankcase and promote rapid breakdown of the oil to form acids which attack the copper-lead bearings used in the test. Typical Chevrolet bearing shells after L-4 tests are shown in Fig. 2. For an oil to pass

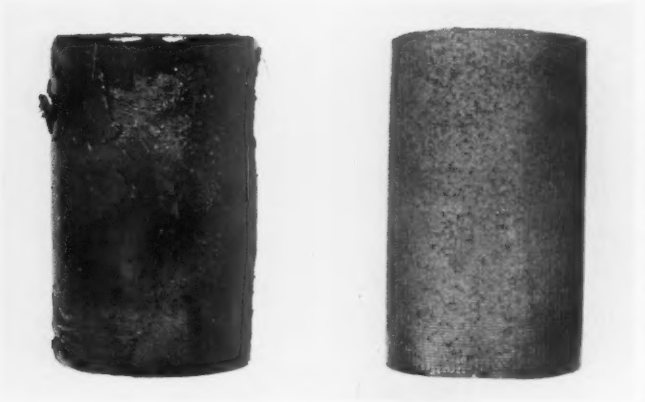


Fig. 5. Filter condition
Left : Heavily sludged after a test on a poor quality oil
Right : After a test on a highly detergent oil

the test the weight loss of the copper-lead bearings should not exceed by more than a small amount that which would be expected from normal mechanical wear; this provides an adequate safeguard against bearing corrosion in practice even in the extreme of heavy duty in tropical climates. It should be noted, however, that oils with anti-corrosive properties are necessary only when bearings of corrodible metal such as copper-lead are fitted to the engine, and even then only when the sump temperature is likely to exceed 250 deg F. Oils for use only in temperate climates need not, therefore, satisfy the corrosion test.

Anti-corrosion properties depend on the quality of the base oil used and the presence of "anti-oxidant" additives. These additives operate either by inhibiting the oxidation of the lubricating oil, or by providing a corrosion-resisting film on the bearing, or otherwise passivating the metal surface. Many attempts have been made to devise laboratory tests which predict corrosion in engines, but no satisfactory correlation of results has so far been obtained. Laboratory tests are, therefore, not recommended at present as they are likely to be misleading.

The Chevrolet L-4 test also yields information on the increase in oil viscosity at the end of the test. It is necessary to limit this in order to avoid difficult starting and excessive friction. Although not specified in MIL-O-2104, the author suggests that the increase in absolute viscosity at 210 deg F should be limited to twenty per cent for both Premium and Heavy-duty qualities.

There is another aspect of corrosion which is not a result of oil breakdown but of direct attack by additives it may contain. This occurs at high temperatures and the parts attacked are notably phosphor-bronze bushes, such as are illustrated in Fig. 3, and silver-lined bearings. Phosphor-bronze can usually be changed for an equally suitable material which is not corrodible by additives, but where silver bearings are used it is important that that the additive shall not attack the metal at the maximum working temperature. General Motors Corporation has devised a test for measuring silver bearing corrosion, designated E.M.D.-L.O. 201/47. In this test a specimen cut from an actual bearing is suspended in a beaker containing the oil which is stirred and maintained at a temperature of 300 deg F. After 72 hours the specimen should exhibit only slight discoloration and the change in weight should not exceed 5 mg.

Engine cleanliness

Probably the most important attribute of a lubricating oil after its capacity as a lubricant and coolant of the working parts is its ability to keep the inside of an engine clean and avoid the sticking of piston rings. Internal deposits may arise from the fuel or the lubricating oil or, more probably, a combination of both. The deposits may be hard lacquer-like films on the piston and similar, though heavier, deposits in the ring grooves; or they may be soft sludges, more generally found in the crankcase, rocker boxes and timing cases, particularly on the cooler surfaces. The deposits are accentuated either by excessively hot or excessively cool running, with the more normal operation giving the best condition. Thus, heavy duty on the one hand and "delivery round" duty with frequent starts and stops on the other hand represent the extremes.

It is clear that the lubricating oil, by virtue of its inherent characteristics and the additives it contains, in keeping the engine clean, will delay the need for a major overhaul and thus give a genuine saving in running cost. All oils may inhibit deposit formation in an engine for a certain length of time, after which they become saturated with deposit-forming material and deposition begins[7]. Detergent additives greatly increase the amount of such materials that can be held in the oil and delay the onset of deposition. This is well illustrated by the pistons shown in Fig. 4 and the filters in Fig. 5.

Before any laboratory engine test to measure lubricating oil quality is adopted it is important that correlation with service engines should be shown. In order to do this it is necessary to carry out extensive field tests with a variety of engine designs and lubricating oil qualities. The conditions in the laboratory engine can then be adjusted to give the same relative results on the range of lubricating oils as those obtained by the statistical examination of the results from the field. There are three factors of prime importance in correlating laboratory

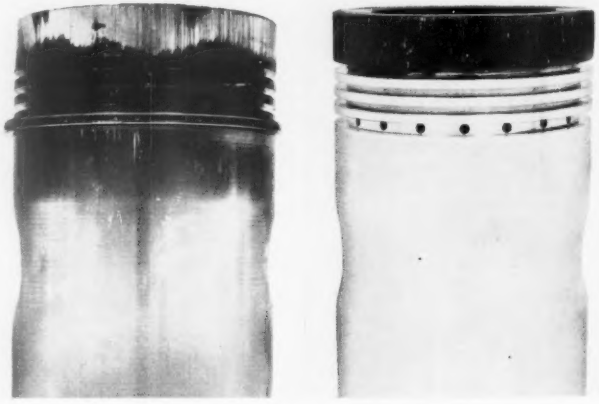


Fig. 6. Caterpillar L-1 pistons—thrust sides
Left : After a test on a Heavy-duty oil of inadequate quality
Right : After a test on a high quality oil to MIL-O-2104 specification

with service engines: 1, the piston temperatures, particularly in the region of the top ring groove, should be closely matched; 2, the fuel qualities should be comparable; and 3, the quantity of fuel consumed in relation to the total quantity of oil reaching the engine by first fill, changes and consumption should be similar to that obtaining in service. This last requirement is particularly important in order to ensure that the oil has sufficient punishment, and failure to take it into proper account is the main reason why short accelerated tests give misleading results.

The first engine test unit to be widely used for measuring Heavy-duty oil detergency characteristics was the single cylinder Caterpillar diesel of 5¼ in bore and 8 in stroke, which was specially designed for the laboratory testing of lubricating oils for ring sticking, and piston deposits. It first came into general use about 1937 because the Caterpillar Tractor Co. would only give approval to branded motor oils which had passed their 480 hr. test in this engine.

The piston and cylinder assembly is essentially that used in the standard production engine, and the general construction of the unit is very rugged so as to minimize the risk of mechanical failures which would interfere with the test routine. Moreover, the unit is so designed that it can be dismantled in twenty minutes for inspection of the cylinder liner, piston and rings. As a test engine it has proved very satisfactory.

In 1941, the U.S. Army Motorized Ground Forces standardized the use of Heavy-duty oils and, under the Allied pooling arrangements, this policy was extended in 1942 to include British military equipment. Since that time Heavy-duty oils to 2-104B specification have been subject to qualification tests laid down by the Co-ordinating Research Council in New York. Originally there were five tests but only two have survived, and these are based on the original tests developed by Caterpillar and General Motors, using the experimental single-cylinder Caterpillar diesel engine (C.R.C. L-1 procedure) and the Chevrolet engine (C.R.C. L-4 procedure) respectively. Both these procedures are at present widely used in Great Britain and the United States.

Fig. 8. Chevrolet FL-2 oil strainer sludging
Left : After a test on a Regular oil
Right : After a test on a detergent Premium quality oil

The more important running conditions for the Caterpillar engine when using the C.R.C. L-1 480 hr procedure are as follows: speed 1,000 r.p.m.; b.m.e.p. 80 lb/in^2; lubricating oil temperature 145-150 deg F; and jacket temperature 175-180 deg F. The top piston ring groove temperature, which is the main factor controlling ring sticking and groove carbon, is then 320 to 365 deg F. The conditions are easy from the point of view of the crankcase but moderate with regard to the piston. Oils are judged by the condition at the end of the test of various parts of the piston, including ring grooves, lands and skirt. An elaborate assessment procedure has been developed which makes it possible to give numerical ratings to the discolorations and deposits. Typical Caterpillar L-1 pistons are shown in Fig. 6.

Recently the U.S. Army 2-104B specification has been superseded by the MIL-O-2104 specification which stipulates that the fuel for the L-1 test shall have a sulphur content of not less than 0·35 per cent instead of not more than 0·4 per cent. Oils pass the L-1 test if, on completion, there is only a small amount of carbon in the top piston ring groove, if there is no more than a trace of lacquer in the second ring groove, and if the piston skirt is free from lacquering.

Further tests have been developed on the single-cylinder Caterpillar engine for testing improved Heavy-duty oils which, before the introduction of the MIL-O-2104 specification, were designated as Supplemental List 1 and 2 oils, the specifications for these improved High-duty oils being supplemental to the original 2-104B specification. However, even though the qualities have been withdrawn from the U.S. Army lists, many oil companies and engine builders recommend their use under the names Series 1 and Series 2 when engines are operated under special conditions.

Series 1 oils are tested by the Caterpillar L-1 procedure with the fuel sulphur content increased to 1·0 per cent because it is claimed that, in this engine, the use of high-sulphur fuels increases engine fouling (Caterpillar 1-E procedure). Series 2 oils are operated on the same high-sulphur fuel under supercharged conditions which further aggravate fouling and ring sticking. The supercharged test is known as the Caterpillar 1-D procedure, and is run for 480 hr at 1,200 r.p.m., 135 lb/in^2 b.m.e.p.,

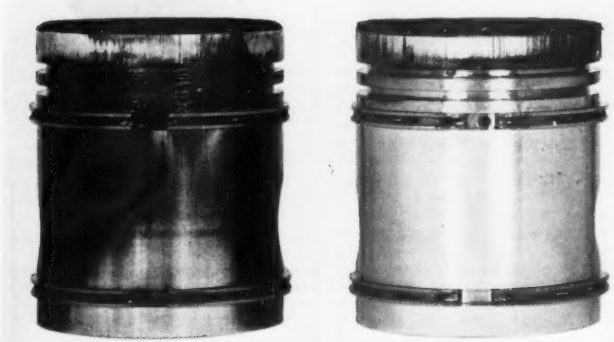


Fig. 7. Sunbury engine piston condition—thrust sides
Left : After 100 hr test on a Regular oil
Right : After 100 hr test on a detergent Premium quality oil

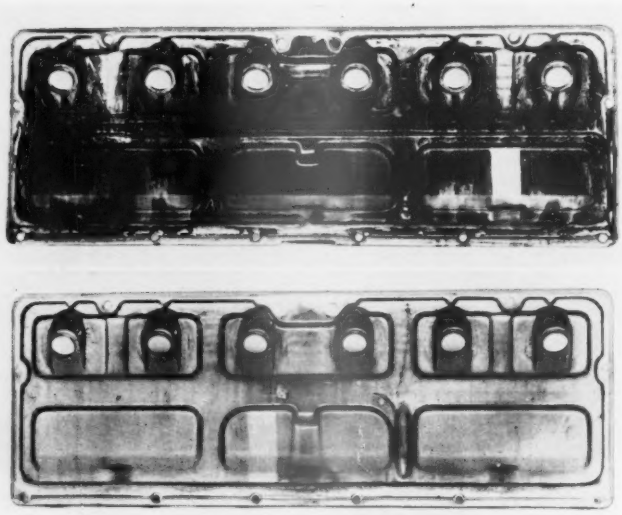


Fig. 9. Chevrolet FL-2 tappet cover sludging
Upper : After a test on a Regular oil
Lower : After a test on a detergent Premium quality oil

175 deg F temperature of the oil to the bearings, and 200 deg F coolant outlet temperature. This test is thus much more severe than the L-1 in all respects.

As far as the Chevrolet L-4 piston condition is concerned, the test does not give a reliable discrimination between oils in either the Premium or Heavy-duty classes.

The above discussion has been concerned mainly with the behaviour of Heavy-duty oils under severe conditions, and it has been shown that there are three recognized standards at present: MIL-O-2104, Series 1 and Series 2. Unfortunately there is no accepted detergency test for Premium quality oils in spark-ignition engines and, consequently, no standards have been set in any general specification. In the author's company a suitable test has been developed on the Sunbury lubricating oil test engine[8,9]. By controlling the piston ring groove temperature to a value which is close to the extreme value in service (355 deg F), a satisfactory test of 100 hours' duration has been established. It is believed that oils are properly graded by this test in relation to their piston cleanliness in service, see Fig. 7.

The low temperature behaviour of oils may be judged by a test such as the Chevrolet FL-2 although some authorities consider that it is not sufficiently extreme as far as cool-running is concerned, and that it is more useful as a fuel test. The running conditions of this test are: speed 2,500 r.p.m.; b.m.e.p. about 65 lb/in^2.; lubricating oil temperature 165 deg F; and jacket outlet temperature 95 deg F. Chevrolet oil strainers and tappet covers after FL-2 tests are shown in Figs. 8 and 9 respectively.

In the author's opinion the proper evaluation of lubricating oils under cool-running intermittent operation can only be made if the test engine is run under repeated cycles of load and speed which closely simulate the practical operation. Such a test has been developed using a 4-cylinder, 2·2 litre, post-war passenger car engine. The running conditions chosen were:—

1 Idle—1 min
2 30 m.p.h. level road load—10 min
3 Idle—½ min
4 Shut-down—18½ min.

This cycle thus takes 30 min to complete. During the shut-down period the engine is cooled by flushing cold water through the jacket and by directing powerful cold water jets on to the sump. The water outlet is 65 deg F at the start of the cycle and reaches a maximum of 150 deg F at the completion of the 10 min at the equivalent of 30 m.p.h. The test equipment is made independent of operators by means of automatic controllers and servo devices. A test consisting of 500 repetitions of the cycle enables satisfactory comparisons of oil quality to be made, and is equivalent to 2,500 miles of low temperature "delivery round" duty.

Heavy sludge deposits are formed by this test, and it has been shown that the weights of sludge can be reduced to half of their usual value by the use of a detergent Premium quality oil. Heavy-duty qualities give even greater improvement. Typical sludge deposits on the cylinder head below the rocker cover are shown in Fig. 10. It is clear that a good detergent additive is capable of giving a cleaner engine not only under high temperature, heavy-duty conditions but also under light-load, low-temperature, intermittent conditions.

Wear

Reboring is one of the major maintenance costs in the life of an engine and reduction in the rate of wear therefore represents an important saving. Wear usually increases with load and speed, and can result from any or all of the following causes:—

1 Abrasion from foreign matter reaching the engine through the air intake, crankcase breather, fuel, or lubricating oil. Wear from this cause is usually independent of operating temperatures

2 Corrosion from acids in the products of combustion. Wear from this cause increases as the operating temperatures are reduced

3 Indirect effects resulting from the presence of deposits. Wear from this cause usually increases with operating temperatures.

The use of suitable filters, together with their proper maintenance, will almost eliminate wear from abrasion. The lubricating oil can also help if it contains an additive which increases film strength so that film rupture by abrasive particles is less likely.

Corrosion may occur during the early stages of warm-up when the cylinder walls are below the dew point of the combustion products. The normal combustion of sulphur in the fuel to sulphur dioxide does not affect the dew point but the presence of sulphur trioxide in the combustion products increases the dew point and makes possible corrosion and increased wear during normal running if the coolant temperatures are low. The remedy is, of course, the use of a radiator thermostat set to a suitable value (160 to 190 deg F).

The addition of an appropriate additive to an oil avoids corrosive wear even in adverse circumstances[10]; in many cases lower than normal wear rates are obtainable by suitable oils in spite of fuel sulphur contents of over one per cent. The mechanism is thought to be the neutralization of sulphuric acid on the cylinder walls by the alkaline characteristics of the additive.

It is clear, also, that by preventing deposits detergent oils maintain good ring action and avoid blow-by, thus lessening, if not eliminating, wear from this cause. The presence of lacquer deposits on the cylinder or piston surface may also increase wear by entraining abrasive particles so that instead of being washed away they continue to do damage. Again, the virtual elimination of such deposits by the use of a suitable oil will avoid wear from this source.

It follows that carefully conducted wear tests of extended duration, both in the laboratory and in the field, are necessary to develop and prove new products to ensure that the lowest rates

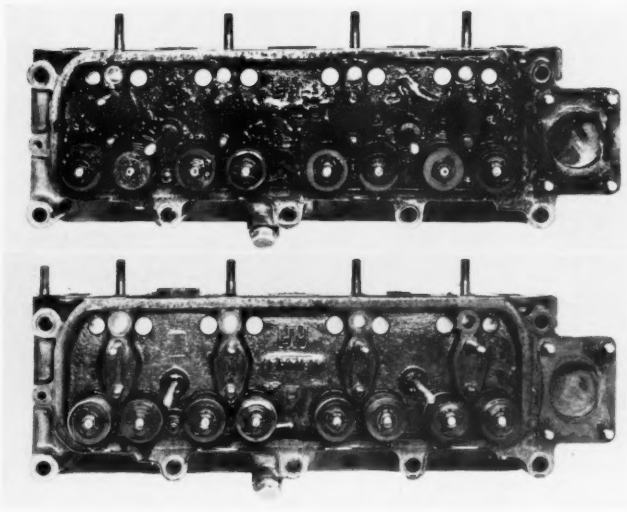


Fig. 10. Cylinder heads from 2·2 litre engine
Upper : After a test on a Regular oil
Lower : After a test on a detergent Premium quality oil

of wear are obtained in service even under adverse conditions.

General conclusions

The lowest possible viscosity is required under starting conditions to provide the fastest cranking speed with the minimum drain on the battery. Under running conditions viscosity is not critical; the grade chosen for an engine should have the lowest viscosity that does not give excessive consumption; reduced friction and improved fuel consumption will then be obtained. Where corrodible bearings are used with sump temperatures in excess of 250 deg F it is important that the oil should reach the standard set by the MIL-O-2104 specification as far as the Chevrolet L-4 test is concerned.

Detergent Premium and Heavy-duty oils are very effective in reducing deposits and wear. It must be pointed out, however, that their qualities are not everlasting and that regular changes at appropriate intervals are necessary to obtain the full benefits. Even Regular oils do not give deposits until a certain concentration of lacquer forming material in the oil is reached. Unfortunately, the time for this to occur is usually too short, and intolerably frequent changes would be needed to avoid deposits and wear by oil change only. Detergent additives in the lubricating oil increase the amount of deposit-forming material that can be held in the oil and increase the running time before saturation occurs and deposit formation begins. The oil should be changed as soon as the effect of the additive is exhausted, but this can be determined only by experience. Where troublesome deposits are formed with detergent oils, it is evidence either that an inadequate quality is used or that the oil change period is too long.

The author wishes to thank the Chairman of the Anglo-Iranian Oil Co. Ltd. for permission to publish this article which has resulted from examination of the performance of lubricating oils at the Sunbury Research Station.

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SAFETY FOOTWEAR

A DEMONSTRATION of safety footwear was given in London recently. To show the effectiveness of this footwear, which is said to be capable of preventing 90 to 95 per cent of common foot accidents, a London Transport double-decker bus was run over the toe of a shoe constructed in accordance with British Standard specification 1870:1952. The wearer of the shoe was able to put his foot into and out of it without the slightest difficulty while the wheel of the bus was stationary on the toe cap. The specification was prepared by the British Standards Institution in co-operation with the footwear industry, the principal heavy industries in which foot accidents are common, trades unions, Government departments, the medical profession and the Royal Society for the Prevention of Accidents.

This standard specification covers the manufacture of the entire boot or shoe, the general appearance of which is in no way different from that of normal footwear. A specially tempered steel toe-cap is fitted either internally or externally in the course of manufacture, and sponge-rubber or similar padding is fitted under the tongue, to protect the upper part of the foot. The cap weighs only two ounces, and is designed to withstand the shock of a sixty-pound weight dropped from a height of 2½ ft, or a deadweight of 3 tons.

The British Standards Institution "Kite" mark stamped on a pair of men's safety boots is a guarantee that they have been made in strict accordance with a rigid specification. Every manufacturer licensed to use this mark has to submit samples to the Institution to ensure that they comply with the requirements, and the factory is inspected to see that methods of production, supervision and testing are adequate. The finished boot bears not only the B.S. mark and grade, but also the name or trademark of the maker, so that faulty goods may be traced back to source. Women's footwear is not included; it is usual to limit protection of such shoes to a special toe-cap, under B.S. 953/3. (2030)

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

The Performance of Lubricating Oils

By H. H. Zuidema.

New York: REINHOLD PUBLISHING CORPORATION. Obtainable from Chapman and Hall, 37, Essex Street, London, W.C.2. 1952. 5½ × 9. 479 pp. Price 40s.

Few subjects have made such progress in recent years as lubrication and especially the lubrication of automobile engines. This book deals mainly with automobile lubrication, because it is in the automotive field where most of the lubricant testing has been carried out and where most knowledge concerning practical lubrication has been available.

The book is rightly named, for after chapters dealing with the process of lubrication and the basic principle of rheology as appertaining to lubrication, almost the whole of the remainder of the book is devoted to the performance of oils. It is, therefore, of great value to automobile engineers, for a knowledge of lubrication matters is becoming of increasing importance. The motor vehicle designer should have a ready knowledge of the effects of correct and incorrect lubrication and what he can do to influence it. The maintenance engineer, either in charge of fleets or in charge of garages, must increase his knowledge, since incorrect lubrication can cost money. This may mean not only that serious damage occurs, but that correct lubrication may save fuel and reduce running costs.

The author discusses oxidation and its effects on the lubrication oil which, in turn, affects engine wear. The chemist will probably understand more of this chapter than the maintenance engineer, but both will find plenty to interest them in all sections of the book. He then discusses bearing corrosion, sludge and lacquer deposition, emulsification and foaming, wear and manufacturing methods, in their respective chapters. Emulsification and foaming hardly applies to automotive oils, since it concerns chiefly steam turbine oils and soluble cutting oils, but is an informative chapter to engineers generally. The last chapter briefly outlines the composition and refining of oil base stocks.

There are several criticisms that might be made. The author has very little to say about detergent oils, their advantages, disadvantages or testing. He does, of course, mention their value, but in view of the facts that this is an American book, that the author is a member of the Shell Oil Company Research Laboratories there, and that detergent oils are actually recommended for use by many United States motor car manufacturers, the omission is noticeable. In this country, these oils are not used to as great an extent. Nevertheless, as in the U.S.A., they have proved very beneficial in overcoming the effects of sulphur in diesel fuel, yet the author barely mentions this. No facts are given concerning the testing of these oils, although this is excusable as the book deals principally with performance, that is, field performance.

On engine wear, its causes, effects, and remedies, a volume could be written. Only the briefest insight into the subject can be included in twenty pages of this book, and it would be easy to criticize the omissions. In the space available, however, the sections on the types of wear, measurement of wear and prevention of wear present valuable information to the reader who may pursue the subject, if he wishes, by consulting any of the many references that are appended to this, and to every chapted. It must be remembered that this book is a Monograph, produced by the American Chemical Society, and it does not profess to be a complete and unabridged treatise upon this very wide subject It is written in a style that makes this usually rather dull subject have interest for automobile engineers and designers and indeed, all engineers and chemists who realise its importance and who want to learn something about it.

Suggestions for inclusion of new material in future editions, in addition to more about detergent oils, might be something on the use of very low viscosity motor oils, and some information on testing oils by means of radioactive tracers. This would bring the section on wear up to date.

The Corrosion Resistance of Tin and Tin Alloys

By S. C. Britton, M.A.

London: THE TIN RESEARCH INSTITUTE, Fraser Road, Perivale, Greenford, Middx. 1952. 10 × 6. 77 pp. 3s. 6d.

The protection of metal from rusting and corrosion is of vital importance to export business, and indeed to many other aspects of manufacturing. Tin coatings are widely used for the purpose and this book gives details of much of the research work on corrosion that has taken place at the Tin Research Institute, under the direction of Mr. Britton. It will prove invaluable to a wide circle of manufacturers who use these coatings, or who will have to use them in future.

The first section deals with corrosion and includes indoor and outdoor atmospheric corrosion, acid corrosion, alkali corrosion, corrosion by milk products, by sulphur compounds, and by non-aqueous media. Notes are included in this section on the effects of impurities in the tin. The second section covers the alloys of tin and includes solders, bearing metals and bronzes. Probably the third section will prove of most value to the practical reader; it deals with coatings of tin and tin alloys. It includes, to quote typical examples, details concerning tinned steel containers for products of low acidity, tin and tin alloy coatings on steel as a preparation for painting, choice of coating weights, tin coatings on lead, aluminium and other metals. Brief descriptions of how the coatings are produced are given and experiences with canned foods figure many times in the text.

It is worth recording, incidentally, that this Institute welcomes enquiries concerning tin and no charge is made for information and assistance. It is financed by tin producers in the Belgian Congo, Bolivia, Indo-China, Indonesia, Malaya and Nigeria and is controlled by the International Tin Research and Development Council consisting of delegates from these six tin-producing countries.

Automotive Maintenance and Trouble Shooting

Edited by Irving Frazee and Earl L. Bedell.

Chicago: AMERICAN TECHNICAL SOCIETY. Available from the Technical Press Ltd., Gloucester Road, Kingston Hill, Surrey. 1952. 5½ × 8¼. 414 pp. Price 38s.

Of the many books dealing with every aspect of automobile maintenance, it is doubtful if there is a single one that could be classed specifically as being devoted to trouble shooting as distinct from overhauling and repairing, or from the technical side of the business. This book deals with every known type of trouble and tells the reader, step by step, what to do about it. It does this in a systematic and practical manner. For a typical example, selecting at random the chapter dealing with the electrical system, we find a section of this devoted to telling you what to do for "Battery low in charge." After recounting how batteries can become low in charge and advising recharging, testing and checking generator output, it is presumed that the output is less than the rated amperes. We then get a "proceed as follows" explanation of what to do next. In the same theme, this is followed by the method of attack when "Generator output low" is the trouble.

A few examples of the many headings, each of which is sub-divided as many times as is necessary to solve any problem, should provide evidence of the scope and approach of the book: Cranks but does not start, Excessive oil consumption, Fuel mixture too lean, Check primary circuit resistance, Test condenser, Intermittent spark at all spark plug wires, Engine overheats and Flush cooling system. Separate chapters are devoted respectively to the electrical system, gasoline engines, gasoline fuel systems, battery ignition, magneto ignition, cooling system, instruments, steering conrol, tyre wear and brakes.

The book deals only with motor cars and does not include commercial vehicles nor diesel engines. It is written by Americans for Americans, but will nevertheless be of value to all motor mechanics. Although the reader may not follow out the procedures given in the exact order suggested or, indeed, may never carry them out at all, the method behind the step-by-step advice and the reason the steps are given in that order will prove distinctly instructive to any intelligent mechanic.

The book was produced by the American Technical Society who have had the services of an Editorial Advisory Board, and the chief editors have had the assistance of several specialized experts in writing most chapters. The reader must bear in mind that detailed information on overhauling a chassis is not given, he will not find how to fit a big end bearing or even how to clean a sparking plug, but he will be told when this work should be done if the motor develops certain symptoms. It should assist him towards logical thinking and ensure that he does not overlook the simple remedies for a fault before committing himself to major repair work which may be avoided. The book is valuable to both the automobile mechanic and to the classroom student.

MACHINE TOOLS

A Review of Trends Disclosed at the Recent Exhibition

ALTHOUGH the exhibition organized by the Machine Tool Trades Association is not the largest of its kind, it is certainly by far the most International. For example, at the recently concluded exhibition there were machines from every machine tool producing country other than those under Russian influence. Slightly more than half of the machines were of British manufacture, and it is worth noting that the only foreign machines for which there were no British counterparts were the Gleason bevel gear generators and Revacycle machines.

Of the foreign countries represented, the U.S.A., Germany and Switzerland were the most important in point of numbers of exhibits. In general, they were also the most competitive for quality in comparison with the British machines. Chief interest in the foreign machines centred on those from Germany. This is understandable, since the German machine tool industry had a high reputation before the war, and this exhibition afforded many British engineers their first opportunity of examining a fairly wide range of German machine tools.

In the four years that have elapsed since the previous International Machine Tool Exhibition, the main apparent developments have been the greatly increased number of copying lathes and the much wider use of electronic controls. Not so apparent, but perhaps of even greater importance have been the developments in metal removing capacity and in accuracy in all types of machines. In fact, some of the machines are designed for use at speeds and feeds that would have been thought fantastic only a few years ago. Furthermore, even with the high rates of metal removal, a very high degree of surface finish is generated owing to the increased power and rigidity of the present-day machines. There have also been advances in designing machines that require a minimum of operational skill even for the production of work to very fine limits.

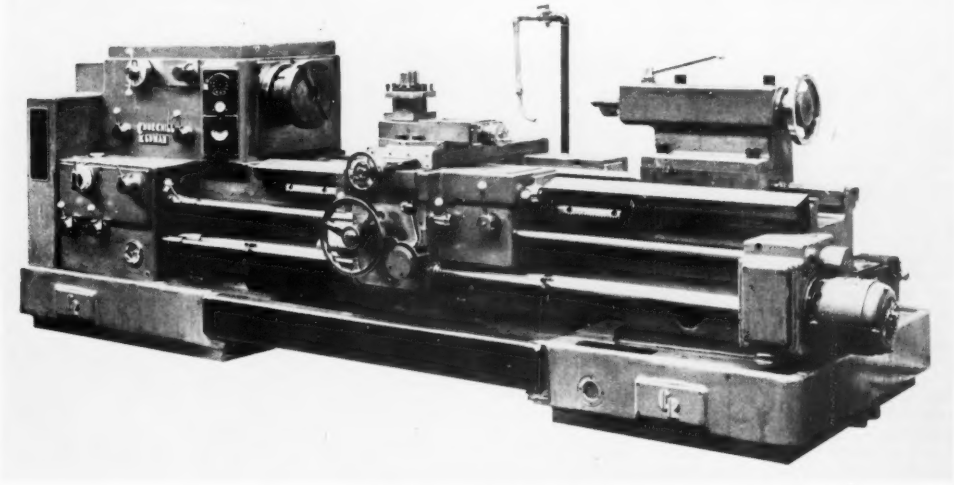
As was only to be expected, many of the machines were examples of well-established designs, but even with these there were many displays of tooling developments of great interest. That so many machines are shown in operation is, of course, a feature of this exhibition that gives a much more accurate conception of what the machine can accomplish than can be provided by a mere description.

Lathes

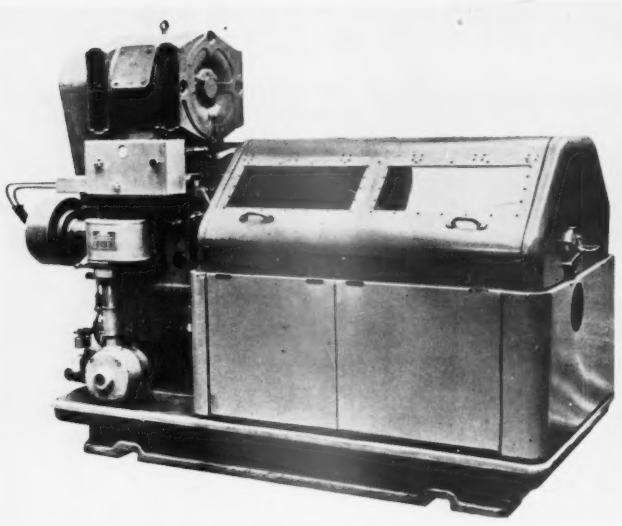
The range of lathes exhibited was so wide, and for many of the exhibits the standard of quality was so high, that any selection must be arbitrary and machines of outstanding quality must either be omitted or dismissed in very few words. For example, the Churchill-Fay 12 in H.V. automatic lathe is a machine with the most remarkable metal removing capacity. This was effectively demonstrated by some machining carried out at the Exhibition on a bull pinion shaft made from a nickel molybdenum steel forging. The component was 10·71 in long, and had four main diameters, two undercuts and a shoulder; the major diameter was 4 in. Despite the toughness of the material a spindle speed of 917 r.p.m. was employed. This gave a maximum surface cutting speed of 950 ft/min with a feed of 0·014 in. The component was completely machined in two operations, each taking 20 seconds.

Of the new lathes to be shown, attention may be drawn to the Churchill-Redman 22L model (*Charles Churchill and Co., Ltd., Birmingham*). This machine has pleasing lines and the bed is rigidly constructed and ribbed to resist deflection. The bedways are widely spaced, the front shear being of the unequal vee type, and the rear shear, which locates the tailstock, is a prismatic vee. The tail and leg also form a tank for coolant. A motor driven coolant pump is built into the tank.

The headstock is arranged to give 18 speeds forward and 6 reverse speeds, with a forward range from 15 to 1000 r.p.m. Transmission is by means of a hydraulically-operated clutch. Braking for the spindle is also provided. The control panel is mounted on the front of the headstock and contains ammeter, tachometer, oil pressure gauge, and push button controls for the main drive motor. A totally-enclosed and pressure lubricated gearbox is employed. It is driven by the end train, which is also positively



Churchill-Redman 22 in × 60 in lathe



Churchill-Fay 12 in high velocity multi-tool automatic

lubricated and totally enclosed. Through the gearbox 30 changes can be obtained, and for additional coarse pitches a 4:1 step up ratio can be provided. The controls for the feed and feed reverse are located on the gearbox. With standard change wheels the gearbox range can be halved or doubled, and with additional change wheels, metric, module and diametral pitches can be cut.

A double wall construction apron is fitted. It has all the controls for sliding, surfacing and screwcutting effectively interlocked. At the front a single lever controls "feed engagement," "quick power forward" and "quick power reverse" to the saddle. The cross slide can also be power operated in either direction. All the mechanism inside the apron is pump lubricated; the same system provides oil between the saddle and the bed-ways. The tailstock has a 4 in diameter spindle and can be fitted with either a solid centre or a built-in revolving centre. Provision is also made for positioning the tailstock along the bed by power operation.

This machine will admit 60 in between centres and has a swing over the saddle of 13½ in. Without changing the end train it has 30 sliding feeds from 0·005 in to 0·038 in and 30 surfacing speeds from 0·0025 in to 0·019 in. Also without any change of the end train it will cut 30 threads from 2½ to 19 T.P.I. The quick power traverse of the carriage is at 12 ft per minute.

Multi-spindle automatics

Of the new multi-spindle automatic machines that were exhibited, the Wickman eight-spindle machine and the four and six-spindle Schütte machines appeared to be those most worthy of attention. All the available data for the Wickman machine was given in the preview of the Exhibition published in the September 1952 issue of this journal. It will be dealt with at greater length at a later date.

The Schütte four and six spindle machines embody several new features. They have been designed to give ample room for fitting the tools and for general accessibility. Two camshafts are provided. They are mounted along the sides of the machines. An interesting point is that ordinary cams have been dispensed with. The special cams provided are fitted with cam levers easily adjustable from the sides of the machine.

For the first time, each longitudinal tool slide is fitted with its own independent feed, which allows the best possible feed to be selected for each spindle operation. It also reduces the need for special attachments. The machines have been so designed that on the four-spindle a 5th and 6th cross slide, and on the six-spindle a 7th and 8th cross slide can be fitted. Both machines are also provided with an automatic spindle stopping mechanism, which comes into operation when the end of the bar is too short to produce a complete component. This greatly reduces the danger of broken tools.

Although many of the other multi-spindle automatics were of established and well-known designs, they showed some very interesting tooling arrangements. One of the most interesting was a six-spindle Conomatic. This machine was shown producing a short hollow cylinder from bar stock with a bore diameter so large in relation to the outer diameter that swarf could be considered the main production. To reduce the wastage, the machine was specially tooled to allow another component to be trepanned from the middle of the bar before the boring started. As a result, the labour and material charges for the extra component were nil.

Single spindle automatics

Many different types and many different sizes of British, American and Continental single-spindle automatics were exhibited. Most of them were established designs, many of which have been modified to give faster production rates. Of the new designs, particular attention may be drawn to the C.V.A. Speed 8 (*E. H. Jones Ltd., 10, Dover St., London, W.1.*). This machine is driven by a reversible motor, enclosed within the cabinet, which drives the primary shaft of the gearbox through a short lightly loaded roller chain. The gearbox provides 39 work spindle speeds in 26 sets of two auto-changes with the pairs in common or opposite directions, for a speed range from 200 to 7,700 r.p.m. Two electro-magnetic clutches in the gearbox provide the two auto-change spindle speeds.

The work spindle is very short and sturdy. It is mounted on twin preloaded bearings front and rear. A special ball-thrust design collet sleeve operates close to the front bearing and dispenses with the usual lengthy push tubes and toggles. An electro-magnetically operated multi-plate, clutch-type brake holds the work spindle stationary when necessary.

An additional vertical cross slide is mounted above the spindle nose. It is operated by one, or by two, cams on the front camshaft. This slide can act as a normal third slide or as an overhead turning slide incorporating, when desired, a simple and most effective provision for profile turning. The swinging bar stop can be used in the normal manner, but there is also provision for feeding to the turret stop and the swing stop successively.

The chucking and bar-feeding mechanism incorporates an automatic self-restoring safety device that prevents damage through over-feeding or feed obstruction. There is also an over-load safety clutch that operates a limit switch to effect the "opening" of the electro-magnetic backshaft clutch. This protects the camshafts from excessive loads due to any cause. A feature of this device is that the angular relationship between the backshaft and the camshafts is automatically restored when the overload is removed. The camshafts, the backshaft and the associated mechanisms are enclosed, yet readily accessible for setting.

The turret camshaft is disposed transversely at the right-hand end of the bed. It carries the disc cam for turret slide feed and a second cam for control of the independent indexing. Two cams for the rotation and feed of the transferring attachment are mounted respectively on and close to the adjoining ends of the front and turret camshafts.

Turret indexing and the associated related withdrawal and advance slide movements are effected by associated, but separate, mechanisms. These mechanisms are mounted wholly on

the bed of the machine, thus relieving the turret slide of considerable weight. This construction also gives a valuable increase in the tool clearing stroke. Moreover, the turret may be both single and double indexed in the same cycle, or alternatively, the slide may reciprocate without any indexing. This machine takes round bar stock up to ½ in diameter, has a maximum feed length of 2 in, and 66 cycle times ranging from ¼ second to 64 seconds.

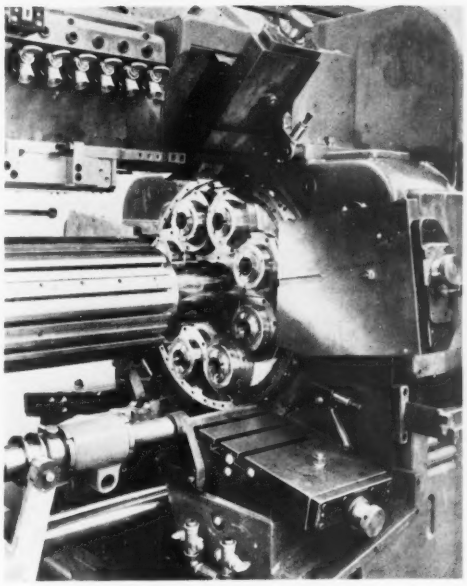
Shaping machines

Although many excellent shaping machines were exhibited, space limitations allow reference to be made only to two. One of these is a new Churchill-Redman 24 in stroke machine (*Charles Churchill and Co. Ltd., South Yardley, Birmingham*). This was a completely new design. The drive is transmitted from a heavy machine pulley, which acts as a flywheel, through a multi-disc clutch and brake. Eight separate speeds are available. The ram is located in 45 deg vee ways which have widely spaced bearing surfaces to give added strength. A star handwheel and clamp allow the ram stroke to be positioned to suit the work. The conventional open throat has been avoided at the front of the ram ways, thus obviating the possibility that chips will enter the body. A direct reading dial indicates the length of the ram stroke.

The tool slide fits into the swivel. It has a micrometer dial, and there is also a split nut for eliminating backlash in the tool slide screws. To relieve the swivel pin of all strain, the thrust of the cutting tool is taken on a solid taper face. The cross slide is normally locked to the vertical ways of the body, and the square slide has four opposed taper gibs for the maintenance of adjustment.

Power feed is provided for the square slide. There are 11 feed changes in either direction from 0·010 in to 0·170 in. These are controlled by a single lever and are registered on a direct reading dial. The feed is distributed over the entire return stroke of the ram, and feeds can be changed while the machine is running. Rapid power traverse in either direction is also provided for the square slide. It can be operated at any time and always in the direction opposite to the feed engaged. Vertical rapid power operation to the cross slide is also provided. This simplifies height settings for different components. Operation of this motion is through the same controls as for the power motion to the square slide. A built - in safety device ensures that damage will not be caused by an attempt to operate when the cross slide is locked.

The Gack punch shaping machine (*Vaughan Associates Ltd., Queen Street, Curzon Street, London, W.*1) was an outstanding example of German machine tool design. This has been developed for precision machining flat surfaces and contours of any radius and shape within the machine capacity. It is also suitable for shaping punches with rounded necks, which hitherto were produced either by costly milling operations or on expensive and complicated machines.



Tooling area on the Wickman eight-spindle automatic

This machine is simple, functional and compact. All controls and other important elements are incorporated in the design and completely enclosed. The work table is vertically adjustable and has a robust horizontal rest. In addition, there is a support at the front that makes cushioning of the table unnecessary. The table top can be swung through 5 deg to both sides. Provision is also made for adjusting the work table both vertically and horizontally. To facilitate setting there are easily readable scales, the vertical graduated to 0·001 in and the horizontal to 0·002 in.

The sliding toolhead is mounted on a flat ram sliding on a vertical surface. A swivel shaft is incorporated at right angles to the sliding surface. It carries the interchangeable arbors for the contour shaping attachment. The mechanism for curving punch necks is housed in the box-type bed. It is connected by rack and pinion to a shaft that controls the tilting motion. Both the tilting angle and the coupling sleeve can be easily adjusted.

A rocker arm gives the tool head its reciprocating motion. As this arm is of the round bar type, it has been possible to obtain a robust drive unit that occupies little space. The conventional slide block has been replaced by a simple sliding member comprising a pin and hole. Length of stroke is easily adjusted from zero to 5·90 in. The universal clamping device comprises a dividing attachment, footstock, co-ordinate chuck and collet holder. Every component can be clamped in the most convenient manner, and the work can always be so clamped that

Wickman 2⅜ six-spindle multi automatic

the entire surface can be completed in one setting. It is possible to make adjustments with the aid of scales, to perform plain and differential indexing, and to turn the workpiece and move it along ordinate and abscissa by the operation of two fine adjustment screws.

A radii adjusting attachment, with a magnifying glass, serves to ensure accurate work location, or to move it during operation, if fixed points, generally centres or radii, must be brought into alignment with the axis of rotation of the dividing head. This attachment has aiming circles for determining and locating radii centres lying outside the workpiece.

Presses

Two noteworthy examples of large British presses were the Forgemaster 1500-ton forging press (*Wilkins and Mitchell Ltd., Darlaston, Staffs.*) and a Vickers - Armstrongs double - action hydraulic press (*Rockwell Machine Tool Co. Ltd., Edgware Road, London, N.W.2*). The Forgemaster has a ring frame and is of fabricated construction. Two apertures in the frame plates give access to the tools. Drive is taken from a motor on top of the press through vee belts to a flywheel, and then through an air-operated multiplate direct action friction clutch to a pinion shaft. The pinions drive two pairs of bull wheels with eccentrics for two connecting rods which are connected at their lower ends to the slides. Two compressed air cylinders, mounted one at each side of the press, balance the mass of the slide and other reciprocating parts. They also take up gear backlash and prevent hammer when the tool contacts the work. Provision is made for ejection of the work from both the upper and lower dies. A strain gauge load indicator is fitted to show the tonnage exerted during the working stroke. It incorporates a switch to cut out the main motor in case of excessive loading.

The British Clearing press is designed as a double-acting press with a punch pressure of 150 tons and a blank-holder pressure of 75 tons. It can, however, be adapted for single-action in such a manner that the full pressure of 225 tons can still be used. Both slides of this machine descend at 450 in per minute under gravity, but the drive is so arranged that the punching speed of 77 in per minute comes into operation when the upper tool is just clear of the blank. The point of speed change is adjustable. At the completion of the working stroke, both slides return at 400 in per minute.

One of the most interesting of the Continental presses was a Schüler 40-ton automatic transfer press (*Pearson Phillips Ltd., Finchley Road, London, N.W.3*). This machine has a frame of tie-bar construction. Drive is taken from a 7½ h.p. motor to both ends of two crankshafts running in phosphor bronze bearings. The drive is transmitted through an air-operated, multi-disc clutch and then by double reduction gearing to give operating speeds of 45, 36 and 26 strokes per minute according to the travel of the ram. The machine can have 10, 12 or 14 tool stations.

The upper tool holders are fastened to the ram by clamping plates. They can be adjusted vertically independently of each other. Vertically-adjustable ejectors are fitted to the punch holders, and the drawing dies can be provided with air cushioning devices. The parts are formed from strip material which is passed from the front to the back of the press by way of a five-roll levelling attachment.

At the first station the blank is cut off and then moved clear of the tool by a slide. Transfer of the work from one station to the next is effected by means of reciprocating rails with interchangeable gripper jaws. The stroke of these rails can be varied from zero to 6½ in. They are driven through a link system that gives simple harmonic motion. At the start of the transfer cycle, the rails are automatically moved towards each other so that the jaws firmly grip the parts. The opening and closing movement can be adjusted to suit the part.

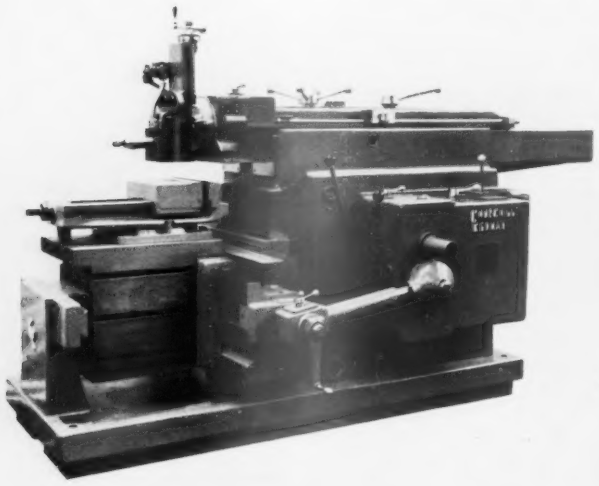
Copying lathes

Many different types of copying and profiling lathes were exhibited. Until recently, Continental and American machine tool manufacturers showed much more interest in the development of this type of machine, but there are now a number of British machines which compare favourably with those of any other country. Of the British made copying lathes, one of the most interesting was the Maxicut electronic lathe (*Drummond Bros. Ltd., Guildford*). This machine is of the same heavy construction as the standard Maximatic lathes. Every slide has its own independent motor, which is electronically controlled to give an infinitely variable range of feeds. There are two models, one admitting 42 in and the other 78 in between centres; each has 18 in swing over the bed and 12 in over the saddle.

Either a single-speed or a two-speed main motor may be fitted. With a two-speed motor, the headstock gives 24 speed changes in the range 100 to 2100 r.p.m. When a single-speed motor is fitted there are 12 speed changes, either from 100 to 1000 r.p.m. or from 200 to 2100 r.p.m. according to the motor speed. A 20 h.p. main motor is generally fitted, but if the machine is intended for heavy duty work, a single-speed 40 h.p. motor may be used. The spindle is mounted in parallel roller bearings which have micrometer adjustment for controlling pre-loading. There are no clutches in the headstock mechanism, the motor being started and stopped by push button control and electrically braked.

The saddle is mounted on the front shear of the bed. It is operated through a leadscrew driven from an electronically-controlled motor in the feedbox at the end of the machine. Provision is made for automatic lubrication, and to facilitate loading and unloading the saddle can be traversed past the tailstock. The cross slide, with its independent drive, is mounted on the saddle. It carries the tracer head and has a maximum cross traverse of 3 in. The cross slide also carries an arrangement of stops to allow six cuts of different depths to be taken along the workpiece. These stops are intended to be used for roughing cuts before copy turning.

A heavy duty rear slide is mounted



Churchill-Redman 24 in shaping machine

on the rear face of the bed. It is independent of the saddle, and can be used for roughing out square shoulders of forgings to save the front copying tool, and for putting in square ruts which cannot be profile turned. The rear slide is adjustable to any position along the bed by rack and pinion motion. It has a maximum feed movement of 3 in. Several tools can be mounted on one rear slide, or if necessary, two or more slides can be mounted at suitable points along the bed.

Feed control for the saddle and the rear slide is mounted in a box at the end of the lathe. Each slide is driven by its own motor. The motors are electronically-controlled and drive to their respective slides through magnetic clutches. Individual operation of the saddle and the rear slides is effected through removable switches operated by adjustable trips on two revolving drums.

The feed control unit has been designed for rapid setting and change-over. Steplessly variable feeds are provided for all slides. For the saddle they range from 1 in to 21 in per minute, and for the cross slide and rear slide from $\frac{1}{2}$ in to 13 in per minute. The saddle has rapid traverse at 100 in per minute, while the cross and rear slides have rapid traverse at 20 in per minute. The piece to be copied is mounted on centres on the front of the machine. Either the part itself or a flat template may be used as a pattern. The centres and template can be adjusted to any position along the bed.

An air-operated tailstock is employed. It is a rigid casting mounted on the back of the bed and carrying a heavy barrel fitted with a revolving spindle mounted on parallel roller bearings. The barrel is air-operated from an air-cylinder built into the rear of the tailstock. Provision is made for easy ejection of the centre, and the tailstock is easily adjustable along the bed by means of rack and pinion motion.

Another extremely interesting British design is the Churchill - Redman hydraulic profiling lathe. This is an hydraulically controlled semi-automatic machine primarily designed for batch production with semi-skilled operation. It is available in two sizes, one admitting 40 in between centres and having 6 in swing over the top slide, and the other admitting 60 in between centres with an 8 in swing over the top slide.

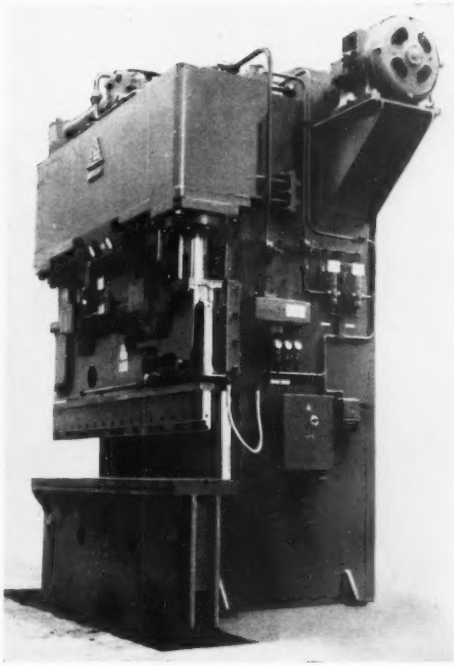
Carriage movements are controlled by hydraulic cylinder and the feeds are steplessly variable from zero to 17 in per minute. The profiling slide is also hydraulically controlled, and is set at 45 deg to facilitate the turning of square shoulders. The standard machine is equipped for hydraulic chucking, and a special compensating chuck has been developed for driving work between centres. A built-in spring-loaded centre, together with end location stop caps, obviates the necessity for controlled depth centring. Hydraulic operation is also employed for the tailstock ram. When slender work is being machined, the operating pressure can be reduced to prevent buckling. Normally, the operation of the chuck and tailstock is sequentially controlled by a pedal, so that the operator has both hands free for loading and unloading, but a hand lever control can be fitted as an alternative.

The tool slide is suitably positioned to allow square shoulders to be turned, and is also arranged to feed into depth automatically. It is also automatically relieved at the end of the cut. The master is held between centres at the rear of the machine, and the tracing finger which engages the master and controls the movement of the tool slide is housed underneath the tool slide. Control of the whole machine is effected through one pedal and one lever. The pedal operates chuck open or close, and tailstock advance or retard; the lever starts the machine cycle. There is automatic knock-off. The complete machine cycle is:—

(1) Depress pedal, tailstock ram advances to work. When the centre in the component is engaged and the component is moved up to its end location, the chuck closes.

(2) Operating lever is moved to "feed" position. The spindle starts revolving and the tool feeds into depth until the tracer finger contacts the master. The carriage then starts to feed at the pre-determined rate, and during the length of carriage movement the cutting tool reproduces the form of the template. Adjustable dogs make it possible to increase or decrease the spindle at any point in the carriage travel. This allows a reasonably constant cutting speed to be maintained on different diameters. When the carriage reaches the end of its travel, the cutting tool is relieved by a pre-set amount, the spindle stops, and in quick traverse the carriage returns to the tailstock end.

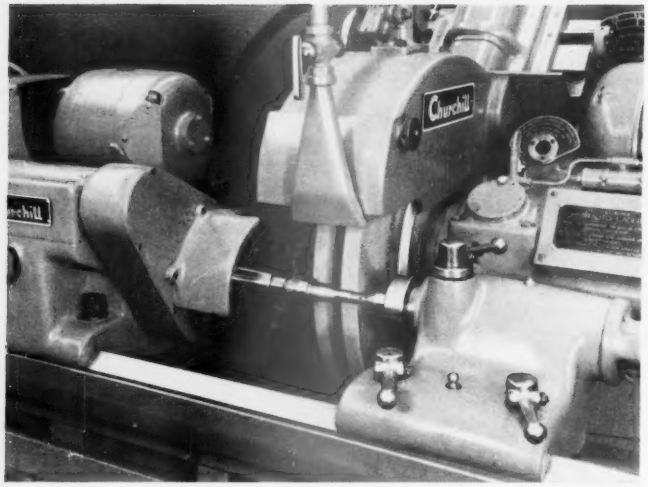
(3) When the carriage return travel is finished, the pedal is depressed to open the chuck and return the tailstock ram so that the work can be unloaded. The machine is arranged for the cutting lubricant to be supplied at the tool point, and the supply is switched on and off automatically with the headstock spindle.



British Clearing double-action press

Three adjustable sizing stops are provided for the cross slide. They allow two roughing and one finishing cut to be taken on a component, with each tool slide position accurately and quickly located. Parallel shafts can be turned without using a template, but the tool in-feed and the tool relief at the end of the cut remain operative. The profiling slide is extremely sensitive and will produce diameters and lengths from the template to within 0·002 in.

Modern German practice in hydraulically controlled copying lathes is illustrated by the Heyligenstaedt duplicating lathe (*Burton Griffiths and Co. Ltd., Marston Green, Birmingham*). On this machine the hydraulic system comprises a high-pressure gear-type pump, a control valve and a servo cylinder. The tracer pin is subjected to a low constant pressure, and a booster element using oil under pressure, transmits to the cutting tool a feed force corresponding to the cutting loads. An extremely low pressure is maintained between the tracer and the template. This allows thin, unhardened sheet templates to be used. The hydraulic cylinder and all the control elements are integrated into a single unit. This arrangement, with the control valve operated by the tracer located within the servo piston, elimin-



Churchill model VW plain grinder with twin grinding wheel

ates oil leaks, hoses and pipe connections. Oil pressure is constantly exerted on both sides of the servo piston so that smooth operation is obtained.

German practice in electronically controlled copying lathes may be briefly illustrated by the VDF "Unicop" machine. On this machine an extremely sensitive tracer transmits minute electrical impulses to an electronic control unit, which controls small magnetically-operated hydraulic valves so that there is no mechanical contact. The high-vacuum valves of the control unit, unaffected by inertia, open and close the hydraulic valves instantaneously at tracer movements of from 0·04 to 0·08 micro-inches.

These valves control feeding pressures measured in terms of tons, and they control them in terms of under 0·01 second and fractions of a milliampere. The ratio between the path of the tracer and the hydraulic valve amounts to 1:1000 maximum, and effects the delivery of a large quantity of oil per time unit for controlling the tool slide. Rising shoulders up to 90 deg and falling shoulders up to 30 deg in the direction of the feed may be machined.

Jig borers

An outstanding machine in its class is the Matrix No. 50 optical jig boring machine (*Coventry Gauge and Tool Co. Ltd.*). This is an excellent example of a co-operative achievement by two organizations, Coventry Gauge and Tool Co. Ltd. being responsible for the development of the mechanical elements, and Hilger and Watts Ltd. (Watts Division) devising the optical system for measuring the table displacements.

Twin wheel forming device on the Churchill VW machine

To ensure the rigidity that is necessary for a jig borer, this machine is of exceptionally robust construction. It can carry extremely heavy work for both boring and milling without risk of distortion or loss of stability. A large diameter quill and spindle are housed in the spindle head which can be rapidly positioned by power. The spindle drive is from a hydraulic motor. It can be transmitted either directly or through back gears. The direct drive is gearless and gives stepless variable speeds up to 2000 r.p.m. to allow fine and high speed boring with cemented carbide or diamond tipped tools. The back gear drive is for fine milling. Provision is made for accurately measuring the quill by the use of slip gauges in conjunction with a built-in dial indicator.

The work table is mounted on compound slides, both of which can be hand or power operated to give an infinitely variable feed traverse for milling. A single position lever, operating in a gate, controls the various feed rates and directions, while two knobs on the front apron control the feed for milling and the speed of the spindle. There is also a handwheel for the final setting or hand operation of the table.

An optical system, specially designed for the machine and constructed by Hilger and Watts Ltd., enables the lateral and transverse movements of the table to be viewed optically on two large illuminated scales. Settings can be made directly in either direction to within 0·0001 in. These readings are projected through a lens system from glass scales attached to the table and the base on to two glass screens, which have optical verniers for fine setting. The complete unit is mounted at the front of the table and is protected from damage by armour plate glass. An important feature is that the glass scales are indestructible and therefore not subject to wear. In addition to the optical system, there are visual external scales on both slide movements for approximate checking and setting. Balanced locks are provided for clamping the slides securely in position without affecting the setting.

The complete table unit has been specially designed to allow coolant to be used during boring and drilling operations. This has been introduced to eliminate the danger that inaccuracies might be caused by local heating of the work. A deep channel surrounding the table connects with the base and the coolant tank. It maintains a free circulation of coolant and at the same time provides an efficient means of chip disposal.

Another example of high class British manufacture was afforded by the 2442 jig boring machine (*Newall Engineering Co. Ltd., Peterborough*). This is a development from their previous models. It retains the open-side con-

struction and the roller system of table location. The new model has increased table movements and a completely redesigned quill and spindle assembly. The spindle runs in special precision double row cylindrical roller bearings. An eight-speed feed box gives feeds from 0·0005 in to 0·012 in per spindle revolution. Other developments are automatic clamping of the head box, which makes it impossible to use the machine with the head unclamped; and rapid approach and withdrawal of the spindle to and from the workpiece, this being controllable to allow rapid approach of the tool to the work with automatic engagement of the selected feed for drilling or boring.

Power traverse is provided for raising or lowering the head box. A depth measuring attachment for the accurate control of the vertical movements of the quill when blind bores are being produced, is incorporated in the head box. Provision is also made for tool ejection. The spindle speeds are steplessly variable between 40 and 2000 r.p.m. through a varionic electrical cabinet. Speed and gear changing are controlled by push buttons. Movement of the table in both directions is effected hydraulically and table setting can be effected directly to 0·0001 in.

An example of German practice for this class of machine was given by the Lindner No. 15 jig borer (*Stedall Machine Tool Co.*, 145, *St. John Street, London, E.C.*1). This machine employs an optical co-ordinate setting system with scales in the form of helices that allow the table to be set to an accuracy of 0·008 mm.

Incorporated in the measuring system are two stainless steel cylinders, one attached to the table and the other to the cross slide. Each cylinder can rotate freely, but is fixed against axial movement in relation to the table or the cross slide. The external surface of each cylinder is ground to a mirror finish and has a fine helical line of 2 mm pitch engraved on it. A micrometer drum is mounted on one end of each cylinder. This in conjunction with a vernier scale allows readings to be taken to an accuracy of 0·001 mm. Built-in lamps to illuminate the cylinders are incorporated in the optical system, which projects magnified images of the helical lines on to separate graticules. These projections are observed through two microscopes mounted on a control unit fixed to the bed.

For moving the table or cross slide through a predetermined distance, a zero setting is first obtained by turning the appropriate micrometer drum until the image of the helical line on the cylinder is located between two hair lines on the corresponding graticule. The table or cross slide is then moved under power traverse through a distance equal to the whole number of millimetres. This travel is measured by reference to scales on the table and the cross slide. The remaining fraction of a millimetre through which the machine member must be moved is then set on the micrometer drum, and final setting is made by means of a fine-feed handwheel which is used to bring the helical line on the scale between the two hair lines on the graticule.

Grinding machines

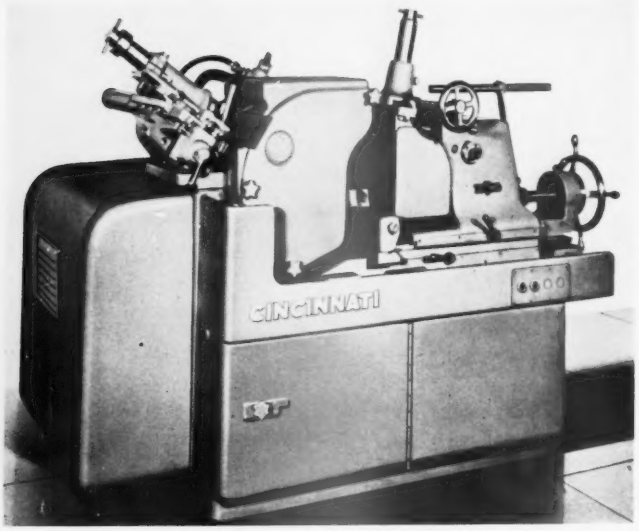
Grinding machines of every type and size were exhibited. In general it can be said that the main features of these machines were the remarkable degree of accuracy that could be maintained consistently, the remarkably high degree of surface finish that can be produced if wheels of the correct grade and grit are used, and the manner in which component accuracy has become a function of the machine.

Cincinnati No. 2 tool and cutter grinder

One interesting exhibit in this class was the model "BW" 10 in × 20 in automatic sizing plain grinder (*The Churchill Machine Tool Co. Ltd., Broadheath, near Manchester*). This machine incorporates hydraulic diminishing feed and an automatic sizing cycle. Normally, it uses a 20 in diameter × 2 in wide grinding wheel, but the machine shown was fitted with twin grinding wheel equipment for grinding two diameters simultaneously. Drive to the spindle is by multi vee ropes from a 10 h.p. motor. The Nitralloy grinding wheel spindle runs in Churchill patent "Hydrauto" bearings which have automatic adjustment and automatic lubrication.

The grinding wheel forming device is permanently in position. It is mounted on the back of the wheelhead. A stepped former bar allows the two grinding wheels to be trued in correct relationship to the work diameters. Traverse of the diamond across the wheel face is by hand-operated screw and nut motion. The upper end of the sleeve carrying the diamond holder is

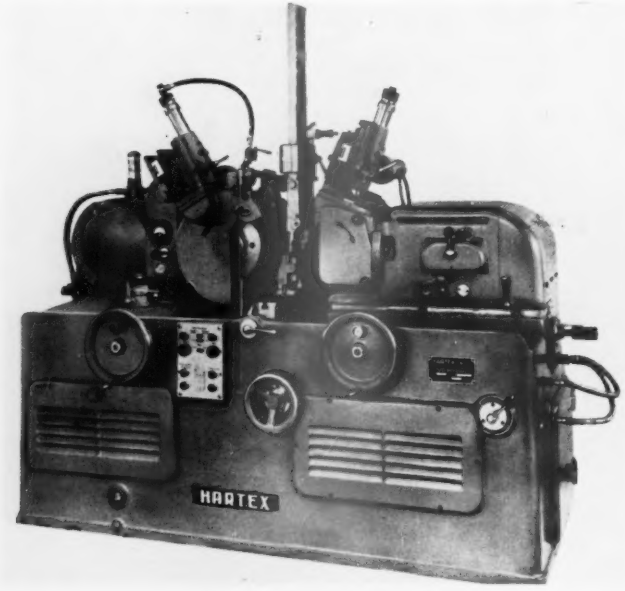
Schütte WU3 tool and cutter grinder



Cincinnati No. 2 centreless grinder

accurately graduated, and the diamond is advanced towards the grinding wheel by micrometer adjustment.

Table traverse is effected hydraulically with an infinitely variable speed range from zero to 192 in per minute. The table speed is controlled by a balanced piston-type hydraulic valve. A 2 h.p. hydraulic pump motor is employed. It drives two geared oil pumps, one for the table traverse and the other for the wheelhead traverse which is run on a separate circuit. The workhead is of the dead centre type and is driven by a variable speed motor with a 6:1 speed range. A two-speed cone pulley is incorporated to give two speed ranges and to maintain the full horse power at slow speeds. The low speed range is from 20 to 120 r.p.m. and the high speed range from 50 to 300 r.p.m. A duplex bush roller chain is used for the final drive to the face plate.



Hartex No. 4 centreless grinder

The Churchill automatic sizing wheelhead on this machine gives a quick forward movement of the wheel followed by a diminishing forward feed movement until the size position is reached. At this point an automatic dwell comes into operation and the wheel is allowed to "spark out" to build up a finish. The period of dwell is variable and may be eliminated completely at the touch of a switch. At the end of the dwell period, the wheelhead runs back at its top speed to the starting position.

Practically all the operating mechanism of the machine is centralized in the control panel and all handwheels and levers are under the immediate control of the operator. The electrical control gear is housed in the body at the rear and is mounted in a withdrawable panel fitted with hinged feet. This greatly facilitates servicing without any dismantling or disconnection.

The "HAC" automatic crankpin grinding machine (*Newall Engineering Co. Ltd., Peterborough*) was of particular interest to production engineers in the automobile industry. This is a well designed machine with all the refinements necessary for ease of operation, high rate of production and extreme accuracy. It has a fully automatic grinding cycle by electrical and hydraulic power. The wheelhead has a 5 in rapid approach with five automatic variable feed rates.

Grinding is carried out in the following manner. The crankpin to be ground is selected and the machine table is positioned by the location of a hydraulically operated plunger in a spacer bar. When this is effected, the contacting shoes of the Newall electro-sizing gauge are placed on the crankpin, and the single operating lever, or joy-stick control, is moved to the operational position. This movement starts the workhead and brings in the wheelhead.

There is then quick approach of the wheelhead to a predetermined position at which the in-feed rate slows while the crankpin webs are ground. Following this, the in-feed rate is again automatically reduced, the coolant starts to flow, and the wheel begins to rough grinds the crankpin diameter. This coarse feed rate continues until the crankpin diameter is from 0·005 in to 0·010 in above finished size, when yet a slower semi-finishing feed is brought into operation and the automatic steady contacts the diameter of the pin being ground. When the crankpin diameter has been ground to within 0·001-0·002 in of finished size, the in-feed rate again decreases to the finish grinding rate, which is maintained until the size is reached. When size is reached, an electrical contact is made in the electro-sizing gauge; this causes the wheelhead to withdraw and at the same time stops the workhead, withdraws the steady and returns the feed dials to the original setting.

Movement of the joy-stick control

away from the operator causes the location plunger to withdraw from the spacer bar, and movement of the control to the right or left causes the work table to traverse in the corresponding direction until a further location is reached relative to a crankpin of the same angular position as the one just ground.

The crankpin is held in throw-blocks by hydraulic clamping. Safety interlocks prevent release of the clamps until the wheelhead is in the retracted position. Should the hydraulic pressure fail or drop for any reason while the wheel is grinding, the clamps remain in the locked position. Both right and left-hand workheads are driven by a large diameter 10-spline shaft and 3 in wide inverted vee chains. Fine adjustment is provided for synchronization.

A D.C. injection braking system stops the workheads in the unloading position. There is provision for "inching." Steadies are fitted at each crankpin position. The steady corresponding with the pin being ground is not brought into contact with the work diameter until the semi-finish in-feed of the wheelhead starts; it is automatically retracted at the end of the grinding cycle.

Automatic precision internal grinding machines are of great importance to the automobile industry, and for this class of machine, advanced British practice is illustrated by the KH Keighley machine (*Newall Group Sales Ltd., Peterborough*). This machine was shown with a high frequency internal grinding spindle for the rapid and accurate production of components with bores in the range from $\frac{1}{8}$ in to 2 in diameter. The outstanding features of this machine are the automatic gauging of the work size, the maximum wheelhead traverse speed of 30 ft per minute, the maximum number of 200 strokes of the wheelhead per minute, and the hand or automatic wheel wear compensation. This machine can also be arranged for profile grinding.

There were several interesting centreless grinding machines of both British and Continental manufacture. One of the machines suitable for a wide range of work was the B.S.A. No. 8 (*Burton, Griffiths and Co. Ltd., Marston Green, Birmingham*). This machine is suitable for work from $\frac{1}{4}$ in to $6\frac{1}{2}$ in diameter. It has self-adjusting bronze bearings for the grinding wheel spindle, and a control wheel driven by a 3 h.p. motor. Either straight-through or shoulder grinding can be carried out. To increase the range of work for which this machine is suitable there are bush type and roller type concentric grinding attachments, and bar grinding equipment is also available. To improve the quality of finish on the work, provision can be made for oscillating the grinding wheel.

An OM No. 2 centreless grinder (*Cincinnati Milling Machines Ltd., Tyburn, Birmingham*), was shown with a Feedmatic hopper capable of feeding parts from $\frac{1}{4}$ in to $1\frac{1}{2}$ in diameter and in lengths from $\frac{1}{2}$ in to 18 in. This machine incorporates a push-button controlled compensating sizing device and a built-in backlash eliminator. This machine also incorporates Filmatic multi-shoe wheel spindle bearings and a one-shot lubrication system for the upper and lower slides. The wheel forming devices are permanently mounted in position, and "crowned" grinding wheel are used for through grinding operations.

Advanced German practice in centreless grinding machine designed was exemplified by the Hartex machines (*Rockwell Machine Tool Co. Ltd., Edgware Road, London, N.W.2*). The B.E.D.4 machine has a capacity of from 0·04 to 2 in diameter. It incorporates rapid withdrawal of the control wheel, a profile wheel crushing device, and the mounting of both the grinding and control wheels on independent slides to facilitate setting.

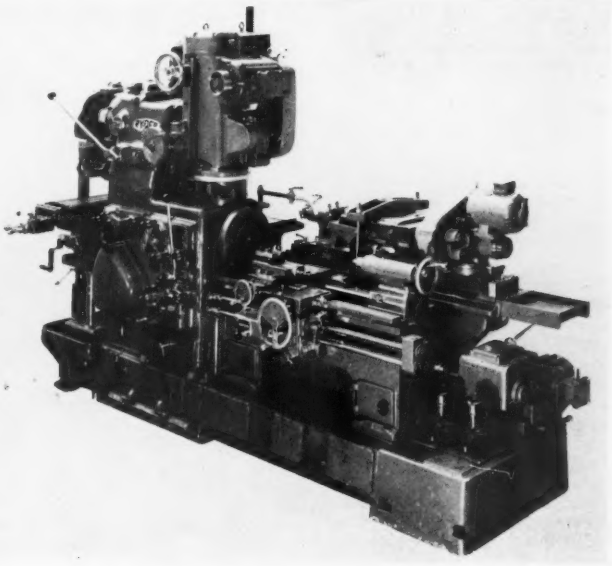
There were many interesting machines developed for toolroom rather than production work. An outstanding example in this class was the Wickman optical profile grinding machine. This is designed to finish grind to fine limits any profile within the machine's capacity. It is particularly suitable for grinding accurately irregular shaped contours on flat or circular form tools, profile gauges, punches and die segments in any materials, including cemented carbides.

In effect, this machine combines the functions of a universal tool room grinding machine, a 50:1 pantograph and a projection unit that provides the optional use of a screen or a microscope as an inspection unit. It is a high precision machine suitable for producing precise forms to a very high standard of accuracy.

A 50-1 drawing of the desired profile is mounted on the drawing table of the machine and is used in conjunction with the pantograph system. The pantograph imparts a 50:1 reduction to its final arm, in which the projection unit is incorporated. Two crossed hair lines can be seen superimposed on the image of the work in the field of the projection unit. The junction of the hair lines follows the path of the pantograph tracing point.

The grinding wheel is carried on a head which is mounted on a series of slides and circular guides. This construction allows the wheel to be positioned in almost any desired angle. For circular work, the wheel may be held stationary, and for straight work it may be reciprocated; then, by manual control of the two traversing slides, the path of the wheel is traversed to the cross hair lines of the projection unit. Irregular shapes and curves can be ground without difficulty.

Many different makes of tool and cutter grinders were exhibited. Two very interesting machines in this class were the No. 2 plain and the Monoset tool and cutter grinders (*Cincinnati Milling Machines Ltd., Tyburn, Birmingham*). The No. 2 machine which was in use for sharpening milling cutters and regrinding centreless work blades used in demonstrating other Cincinnati machines, incorporates an anti-friction table slide and duplicated controls to provide for sensitivity and ease of operation. The universal workhead is arranged to swivel in two planes, and the anti-friction wheel spindle is cartridge mounted. An interesting feature of this machine is



Ryder universal machine from the front

the mist coolant attachment, which is used for wet grinding sintered carbide cutters. Through this attachment a thin stream of coolant is transformed into a mist upon contacting the grinding wheel. Instead of being thrown off the wheel, the mist is sucked back and returned to the coolant tank through a flexible tube. The capacity for using coolant in grinding carbide tools allows production and wheel life to be greatly increased. The Monoset cutter and tool grinder is designed for the production, as well as the sharpening, of tools. Tapered, straight, radius, and radius and ball-nosed cutters can be ground with equal facility.

German practice for tool and cutter grinding was exemplified by Schütte machines (*Rockwell Machine Tool Co. Ltd., Edgware Road, London, N.W.2*). Two universal machines of this make were shown. Both have special journal bearings with automatic oil circulation to ensure clean and accurately ground surfaces. The drive is through a built-in motor without the use of belts. The smaller machine has a hardened and ground ball-track table with a manually-operated movement of 20 in. The larger machine has a hydraulically-operated table movement of 25 in.

One of the most interesting machines in the tool and cutter grinder class was a Walter automatic hydraulically inserted tooth cutter grinder (*Elgar Machine Tool Co. Ltd., Feltham, Middlesex*). The outstanding feature of this machine is that it grinds each blade automatically from the face via the corner or radius to the peripheral edge in one continuous operation so that all grinding can be completed at one setting.

The novel feature of the machine is the grinding wheel spindle, which together with the drive and a short wheel slide, is arranged on a sturdy face plate that moves in slide ways. This construction makes it possible to swivel automatically the entire grinding assembly during the rotation and axial oscillation in such a manner that all blade faces are ground without any resetting of the work. In addition, the machine can be used for specialized grinding operations. For example, closely pitched teeth can be relieved by using a cup wheel which can be located relative to the cutter by swivelling the wheel. For grinding lands on the cutting face, the wheel spindle is turned through 180 deg and a saucer wheel fitted in place of the peripheral wheel.

Special machine

In some ways the most unconventional machine in the exhibition was the Ryder universal machine tool (*A. C. Wickman Ltd., Coventry*). It comprises a lathe, sensitive drilling machine, universal milling machine, universal grinding machine and a tool and twist drill grinder. Originally designed to an Admiralty specification for repair ships, it is compact and rigid and would be useful for general purpose work where there is a lack of floor space.

Both the lathe and the milling machine are powered by one 6 h.p. motor. The drive is transmitted by vee belts to a common shaft from which it separates through multi-plate clutches to the lathe and milling machine spindles through nine identical speed gear trains. These two sets of gears together with the lathe feed screw-cutting gears are housed in the main body of the lathe headstock. The lathe spindle and both the vertical and horizontal spindles of the milling machine are all mounted on identical taper roller bearings. Each spindle has a reverse gear and brake. Stopping and starting are effected through lever operated clutches.

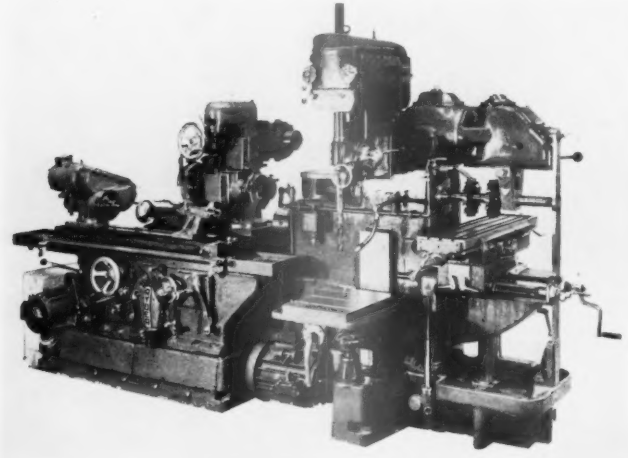
The 6½ in sliding, surfacing and screwcutting lathe admits 30 in between centres and has a swing of 10½ in over the gap bed. There are nine spindle speeds ranging from 21 to 533 r.p.m. and 12 feeds from 0·0056 in to 0·0391 in per revolution sliding and from 0·0028 in to 0·0195 in per revolution surfacing. Change wheels are supplied to allow a wide range of metric and inch threads to be cut. With even change wheels, 12 threads ranging from 3½ to 24 T.P.I. can be cut. The change wheel compartment is on the front of the lathe headstock, but the wheels are readily accessible since their shafts lie at right angles to the lathe spindle.

A vertical and a horizontal spindle are fitted to the universal milling machine. The vertical spindle is mounted in a swivel head that can be moved through any angle. This allows a wide variety of angular milling operations to be carried out without the need for complicated set-ups. A further feature of the head is that it carries the support for the horizontal arbor. The table has a 40 in × 10 in working surface, and 23 in longitudinal, 7 in transverse and 13 in vertical traverse. There are nine spindle speeds from 21 to 533 r.p.m. and four feeds from ½ in to 10 in per minute. The knee of the milling machine is manufactured by Adcock and Shipley Ltd., and is the same as that fitted to their model No. 2AGU machine.

A separate 2 h.p. motor drives the sensitive drilling machine. Changes of speed and feed are obtained through sliding gears, and hand or automatic feed can be used. The spindle and spindle head are counter-balanced by a spiral spring. Both the all-gear head containing the motor and the spindle head are manufactured by Adcock and Shipley Ltd. They are normally fitted to that firm's model No. 24 A.G. The table is elevated by bevels and screw by means of a hand lever. It is self-sustaining in any position. This machine will take drills up to 1¼ in diameter, and has nine speeds from 85 to 1220 r.p.m. and four feeds from 0·0055 in to 0·0162 in per revolution.

The universal grinding machine is suitable for surface grinding and external and internal grinding. It also has a swivel table for taper grinding. A 2 h.p. motor drives the table through a gearbox and clutch. The gearbox is supported on a rubber mounting and is coupled to the grinder by means of a flexible coupling. The wheelhead and workhead are self-contained units driven respectively by 3 h.p. and a ¼ h.p. motor. Both heads are manufactured by the Churchill Machine Tool Co., Ltd.

To prepare the grinding machine for use, the wheelhead adaptor, which is normally clamped to the grinder table, is transferred to the lathe bed where it is coupled to a vertical shaft that transmits the feed motion to the head. The wheelhead can then be lifted from its base on the lathe headstock and mounted on the adaptor base. Therefore, the lathe and the grinder cannot be operated at the same time.



Ryder universal machine from the rear

TAPPET WEAR

Some Development Work Leading to the Adoption of Cast Chromium Alloy Tips

THE Triumph Engineering Co. Ltd., of Coventry, have recently carried out extensive tests in connection with tappet wear. After some initial experiments to determine whether improvements could be made to the tappet lubrication system, the work was aimed principally at discovering better wearing materials for the tappets. It was realized that other lines of approach to the problem, such as the modification of the cam design, were also likely to give satisfactory results. However, the course of action adopted was, in the circumstances, considered to be the most economical, and likely to yield results in the shortest possible time.

A 500 c.c. twin cylinder aircooled engine, with push-rod operated overhead valves was employed for the tests. Because space limitations in this engine precluded the incorporation of rotating tappets, sliding follower type tappets were used. Thus, not only was there a purely sliding interaction between the cams and tappets, but the rubbing velocity was, in general, higher than would be the case with the rotating type. Two springs, positioned concentrically, are employed giving a total spring rate of 158 lb/in. This rate was necessarily high in order to prevent valve bounce within the range of engine running speeds. As a result, the tappet when being swept by the cam flanks and toe was heavily loaded by both the spring and the acceleration forces, but, when riding on the base circle periphery, it carried no more than its own weight plus that of the push rod. The cam and tappet mechanism is normally lubricated by splash from the crankshaft as well as by oil drip which passes from the rocker box through the push rod housings and the tappet guide block.

After preliminary tests, it was realized that engine speed was, in this case, the critical factor determining whether excessive wear took place, or whether the sliding surface of the cam and tappet acquired the polished Beilby layer that is indicative of a long, wear-free life. At speeds up to 2,500 r.p.m. the cams and their tappets wore rapidly; at 3,000 r.p.m. the wear was unpredictable, some pairs being relatively unaffected, and others wearing badly. On the other hand, after 50 hours running at speeds of 3,500 r.p.m. and above, all cams and tappets were in perfect condition. These results were consistent with what was happening in practice. When installed in motor-cycles, engines that were consistently run at high speeds gave no trouble, whereas some of those run at lower speeds wore out their tappets and cams after a relatively short mileage. Accordingly, a fifty hour run at 2,500 r.p.m. was adopted as the standard test.

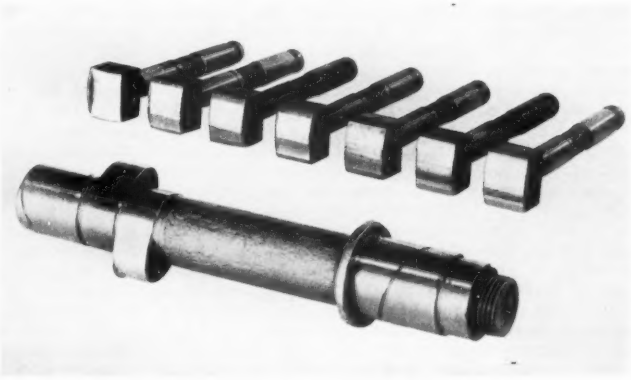


Fig. 1. A few of the tappets tested for wear resistance, and a typical camshaft

One of the first steps in the investigation was to see if wear could be appreciably reduced by improving the lubrication of the rubbing surfaces. With this object in view, the normal splash lubrication was supplemented by a forced feed. A jet of oil was directed between the contacting surfaces, and observed through a window fitted to the crankcase. Several test runs were carried out using different crankcase oils, including a graphited oil and a castor-based racing oil. Also with a view to improving lubrication, many surface treatments were tried on case hardened mild steel tappets and cams. These treatments included grease mopping, phosphate coating with and without graphite impregnation, oxidizing in steam at 200 deg C, shot-blowing, and etching in acids to produce porosity.

It was found early in the investigation that the hardness of the cam and its tappet, the depth of case hardening, and the type of case hardening steel, whether an alloy or straight carbon, have little influence on the wear resistance. There were instances of excessive wear in short running times with both alloy, and straight carbon steels at hardnesses ranging from Rockwell C 58 to C 64 as fitted, with and without retained Austenite in the case structure. Among the materials that wore excessively were tappets made of fully hardened high speed steel, chill-faced cast iron, high carbon chromium ball race steel, and hard chromium facing 0·005 in thick. With these, between 0·005 in to 0·030 in was worn from the cam and tappet sliding surfaces during the 50 hr tests. The chromium facing tended to flake off and, being carried around in the lubricating oil, it heavily scored rubbing surfaces in many different parts of the engine.

Although this did not happen with all chromium-faced tappets, it was difficult to obtain consistently good adhesion between the chromium facing and the base material of the tappet.

Tappets faced with Stellite by using a gas welding technique to run the metal on to the tappet from a Stellite welding rod, showed no appreciable signs of wear after the standard test. Their mating cams, on the other hand, showed signs of wear varying from

slight to severe. Fairly good results were obtained with tappets of En 4CA material, nitrided 0·020 in deep. With these tappets, the cams were only worn 0·002 in.

The wear problem was finally solved by using a cast chromium alloy. This material, so far as is known, has never before been produced in this country. Initially, a welding rod of this alloy was made by Deloro-Stellite Ltd. The metal was run on to the tappets to a thickness of about $\frac{1}{8}$ in. After cooling, the tips were ground to the correct shape and size. Many fifty-hour test runs were made at 2,500 r.p.m. as well as at other speeds, and under all three conditions consistently good results were obtained. Both the tappets and the cams were in perfect condition after the tests, their mating surfaces showing only polished tracks where sliding contact had occurred. Facing with this alloy, now known as Delchrome, was then put on a production basis, and the faced tappets are now used in conjunction with a camshaft made of Kasit.1. steel produced by Clyde Alloy Steel Co. Ltd., case hardened to a depth of 0·030-0·040 in.

Fig. 2. The tray on the left has water flowing through it to keep the stems of the tappets cool

Some of the tappets, after wear resistance tests, together with a typical camshaft, are shown in Fig. 1. The left hand tappet is a case hardened mild steel component after 7,656 miles on the road; the remainder have all been bench tested for fifty hours at 2,500 r.p.m. They are, in order: case hardened mild steel, case hardened mild steel phosphate coated and graphite impregnated, chilled cast iron, high speed steel, case hardened mild steel chrome faced, and a tappet tipped with Delchrome.

Theory of wear

Insufficient is known about the wearing action between cast chromium alloy and case hardened steel to give an explanation for the superior wear resistance. Nevertheless, a few general remarks on wear of metals will perhaps be pertinent, and may stimulate thought on the subject. It is stated, Ref 1, that the wear of solid surfaces is a complex process which under many conditions includes both chemical attack and physical damage, so that any slight change in the operating conditions may alter the whole nature of the wear process.

The area of contact of two metal faces bearing on one another is limited to that of the contacting peaks of surface roughness or asperities. For example, it has been shown by measurement of electrical resistance, Ref 2, that steel flats under a load of 20 kg and with an apparent area of 21 cm^2, have an area of contact of only 10^{-4} times the apparent area. Local pressure at the asperities must therefore be 10^4 times greater than the mean value expressed in terms of the load over the total area.

When the surfaces are sliding together, the frictional work done must be dispersed as heat. In view of the relatively minute volume of each of the peaks in contact, it follows that their temperatures must be very high. In Ref 3, experiments are described which suggest that not only is this maximum temperature very high, but it is limited only by the melting point of the sliding metals. When the metal melts, the peaks of the asperities naturally flow. Moreover, at temperatures approaching the melting point, plastic deformation of the peaks may be expected. Similar results are obtained if lubricant or water is present, since the hot spots are formed so rapidly that the additional cooling agent has little effect, Ref 4.

At these high temperatures a metallic junction or welding occurs, and as the sliding motion continues these junctions must be sheared. The shearing may take place in four different ways. If the junction is weaker than the metals themselves, shearing will occur at the interface. Consequently the amount of metal removed from either surface will be very small. This occurs, for example, with a tin base alloy sliding on steel. Junctions of this relatively weak nature are usually formed when tough films of oxide, sulphide or chloride are present. Even though these films are only of molecular dimensions they may, provided they are not broken up by deformation of the underlying metal, hinder the formation of strong metallic junctions.

If the junction is stronger than one of the metals, shearing will often occur within the bulk of the weaker metal, fragments of which will be left adhering to the harder surface. Under these conditions the amount of material thus removed may be relatively large, even though the friction is similar in degree to that observed in other cases in which little wear occurs. That is to say, there is no definite quantitative relationship between friction and wear. This type of wear results in the gradual build up of a film of the softer metal on the harder one, so that ultimately the sliding is characteristic of that obtained with two surfaces of the softer metal. Consequently, the friction, surface damage, and wear are very high.

Sliding action between similar metals is the third way in which wear may take place. In this case, the process of deformation and welding usually work-hardens the surfaces, and appreciably increases their shear strength. Consequently, shearing rarely occurs at the interface, but takes place within the bulk of the metals, For this reason, the surface damage of both sliding bodies is considerable. On the other hand, a few alloys work-soften; these generally have good wear resistant properties.

The fourth type of wear is that which occurs when the junction is stronger than both of the metals. In this case, most of the shearing will

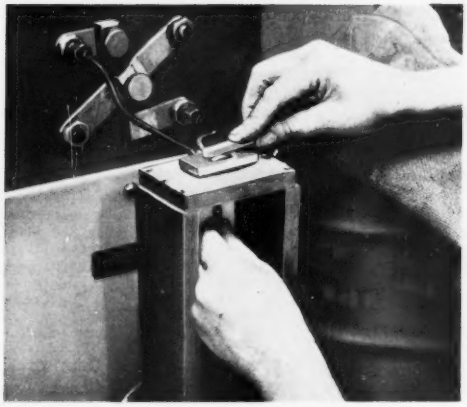


Fig. 3. A square section asbestos rod is used to position the tip as accurately as possible

Fig. 4. Tappets in various stages of manufacture. They are, from left to right: welded-on tip, welded and ground, brazed-on tip, brazed and ground

take place in the weaker metal, which will show the greater signs of wear. Nevertheless, it will also occur to some extent in the stronger one, in which the wear may still be appreciable.

Mailander and Dies, Ref 5, make a clear distinction between *seizure* wear and *abrasive* wear. Seizure wear is said to occur when there is a heavy tearing of the surfaces, and marked vibrations are set up in the moving parts. When similar metals are in rubbing contact it occurs even at the smallest loads, and the wear is always heavy. Abrasive wear is said to occur when the surface damage is much less severe, and the surfaces can run safely until they are completely worn away. However, the distinction between these two types of wear seems to be a matter of degree rather than of kind, Ref 1.

Dies and others have found it convenient to restrict their investigation to the field of abrasive wear. With mild steel sliding on a hard chromium steel disc this involves working at pressures below 30 kg cm^2. Under these conditions, wear is considerably less than for mild steel on mild steel. As the load is increased, it is found that the wear at first becomes more rapid, and then falls, finally rising again. These results suggest that the decrease in wear is due to a transition from one type of wear to another. The transition point is determined largely, though not exclusively, by the surface temperatures developed during sliding.

For small loads, and therefore low temperatures, the wear products are mainly FeO and Fe. Beyond the critical load, however, the amount of Fe in the products becomes negligible, and the major part consists of Fe_2O_3 and FeO. At the same time a significant change occurs in the hardness of the surface layers of the steel slider. Up to the critical load, the surface hardness is somewhat higher than the bulk hardness; beyond it the surface hardness increases appreciably. This marked increase is attributed partly to the formation from the atmospheric nitrogen, of nitrided layers which penetrate to an appreciable depth into the surface, and partly to the work hardening of the metal under the more severe conditions of operation.

Surface oxidation can play an important part in the wear process. Oxide films formed on the rubbing surfaces may, in some cases, function as lubricants. In other cases, hard oxide may be occluded into the rubbing surfaces, the occlusion being moved deeper and deeper, at each motion of the slider, by the deformation of the layers of metal near the surface. The presence of occluded oxide may inhibit grain growth and adhesion between surfaces. Again, if the oxide is very hard and embeds itself firmly into the underlying parent metal, the rubbing face may become, in effect, an abrasive surface. This explains the striking fact that, under conditions favouring oxidation, a soft metal such as tin produces heavier wear on a hard chromium steel disc, than does a hard steel slider on the same disc.

Sir George Beilby, Ref 8, shows that a polished layer resulting from the repeated rubbing of two surfaces is structurally very different from the underlying metal. His view is that the layer is amorphous in character and has properties similar to a highly viscous liquid. It flows over the surface bridging the minute chasms and filling up the irregularities. More recently, it has been suggested, Ref 9, that the polish layer is always microcrystalline, but the crystal size is so small that it appears to be amorphous.

From the results obtained in the tappet wear investigation under consideration, it would appear that room temperature hardness of the metal is of secondary importance. This is shown by the fact that the hardness of the cast chromium alloy, Rockwell C56, is lower than that of electro-deposited chromium, hardened high speed steel, and case hardened mild steel. Nevertheless, an investigation into other aspects such as the melting point of the cast chromium alloy relative to the hardened surface of the camshaft steel, oxide formation and other chemical and physical changes might throw some light upon the reasons for the low rate of wear. However, this is a fundamental metallurgical problem, and it is probably too large an undertaking to be performed by a manufacturing company.

Production

Facing the tappets with a welding rod, by the method used experimentally, was slow, wasteful and expensive. This was largely due to the need for removing excessive amounts of metal by grinding. As a result, a more economical procedure had to be devised. In the method now employed, the alloy is cast, to close limits, in sticks of the necessary D-section. An elastic grinding wheel is then used to part it off in lengths equal to the width of the tappet. The joint face is rough ground, and the tips are then silver-brazed on to a shortened tappet. A small high-frequency induction set is used for this operation.

The arrangement of the equipment on the bench is shown in Fig 2. On starting work, the operator takes an untreated tappet from the pile on the

right, coats the joint face with Easy-flo, stainless steel grade flux, and places it in the rack on the right. A piece of Easy-flo No 3 silver-brazing foil, which is supplied cut to the appropriate shape and size, is placed over the flux on the joint face. Then the joint face of a Delchrome tip, taken from the container in front of the rack, is coated with flux, and the tip is placed over the foil on the tappet. The whole assembly is next positioned in the induction heater coil on the stand in front of the heater set. Under the stem of the tappet, support is furnished by a screw jack adjuster which is used to set the vertical position of the tappet relative to the heating element.

While the heating is taking place, another tappet is prepared in the manner already described and placed in the rack. By the time this has been done, the first prepared tip assembly is at brazing temperature. The tip is positioned as accurately as possible with the aid of a square section asbestos rod, and pressed firmly down. This operation is shown in Fig 3. It can be seen that although the head of the tappet is at brazing temperature, the stem is cool enough to be gripped by hand. An armrest has been introduced to assist the operator in positioning the tip.

The tappet is then removed and placed in the rack on the left. In order to prevent loss of hardness due to heat flowing down from the head, the stems are positioned so that they are submerged in cooling-water flowing through the pan in which the rack is standing. The next tappet, already prepared, is placed in the heater coil, and the whole sequence of operations repeated. Finally, the tappets are taken away in batches to be finish ground, and inspected.

Initially, trouble was experienced with faulty brazing. However, this was rectified by employing the more suitable grade of flux quoted in the description of the production operation. Rejections are now less than 0·5 per cent of a weekly production of 1,500 to 2,000 tappets. The manufacturers state that confidence in the results of the investigation has been amply justified, for during the twelve months that this weekly rate of production has been maintained, not one case of cam or tappet wear in service has been reported.

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MECHANISM CLEANER

DESIGNED by engineers possessing wide experience of the needs of the meter and the instrument industries, the NSE mechanism cleaner is a robustly built machine capable of handling a variety of small mechanical and electrical parts, meter and clock gearing, and precision mechanisms and components.

As the cleaning basket is 8 in diameter it is capable of holding moderately large articles, but removable subdivisions are provided to accommodate small parts. The basket is detachable, and clamps are provided to enable a complete clockwork or other mechanism to be held in position and cleaned as a unit.

The machine has two chromium-plated, tubular steel pillars fitted to a ribbed, cast aluminium base, approximately 3 ft in diameter. Sliding on these pillars is a rigid, cast aluminium head-unit carrying a one-sixth h.p. repulsion-induction motor with push-button starter, fuses and neon indicator lamp.

A hand lever on the brush gear gives a wide speed range without the use of a rheostat. The motor is connected through a reduction gear and a shock-absorbing clutch to a robust, vertical spindle mounted in roller bearings and fitted at its lower end with a faceplate to which the work-basket is secured by thumb-nuts. Additionally, the faceplate is slotted to receive clamps by means of which a complete mechanism may be attached without the work-basket. The clutch permits the faceplate to be rotated by hand in either direction if desired, without damage to gearing.

Of perforated steel, the work-basket is approximately 8 in diameter and 4 in deep and is heavily cadmium-plated. The removable sub-divisions enable a number of different items to be cleaned at one operation and also prevent small articles from being crowded against the wall of the basket. When immersed in the cleansing solution the rotational speed of the basket is about 120 r.p.m.

The NSE mechanism cleaner

The head-unit is counterbalanced by means of two springs contained within the pillars, and a handwheel with a ball thrust bearing enables it to be firmly clamped in any position. An aluminium lid, with a double lip to prevent splash leakage, is attached to the head-unit and fits closely on top of the cleaning tank when the basket is immersed. Centred round one of the pillars is a cast aluminium turntable mounted on rollers, and on this are located three rectangular glass tanks and one ventilated metal drying-box. The turntable is indexed for position, and is located by means of a steel pin. The three tanks holding the cleaning fluids each require approximately one gallon of liquid to half fill.

The metal drying-box stands over a hole in the turntable below which, when the box is in the drying position, is a circular heating element protected by a mica shield. A micro-switch beneath the turntable is arranged to switch on the heating element just before the drying-box comes over it, and to switch off the element when the turntable is moved to the next position. A red neon indicator lamp on the head-unit shows when the heater circuit is switched on.

Manufactured by Northern Scientific Equipment Ltd., Bark Street, Bolton Lancs. and distributed by John Baggs, Technical Services, Uppermill, Oldham, Lancs., the machine is finished in crackle black enamel and chromium plate. Arranged for connection to the A.C. mains it is normally produced for use on 230-240 volt, 50 cycle, single phase mains but machines for other voltages can be supplied to special requirements. Current consumption, including the heater, is approximately 600 watt. (2026)

STARTING IN EXTREME COLD

A Study of the Special Difficulties with Small-capacity Engines

By J. D. Richards*

IT was only after the recent war that the car industry of the United Kingdom entered many world markets on a large scale. A number of important new problems had to be overcome in a short time, to meet road and climatic conditions very different from those for which pre-war models were suitable. One of the most important of these problems was the correct carburation of engines of about two litres capacity and less, over the wide temperature variations found in many different markets.

To what we already knew, we were able to add a great deal in a short time by studying American technology. With carburation, however, there are some very different factors which make too close an emulation dangerous to success. As these differences arise mostly in very cold climates from the fact that our engines are of smaller cylinder capacities, and as the future trend is likely to be more towards engines of under 1000 cm³ capacity, it is important that such points should be given primary consideration if a satisfactory product is to be evolved.

General problem

The general problem can be divided into two parts, it being understood that there is a degree of interaction. These two questions are starting from cold and drive-away.

Starting from cold. Although conditions are encountered where starts have to be made at −20 deg F or below, it is nowadays considered that a safe start at −10 deg F is enough to give a satisfactory performance. Prior to the war, +20 deg F was taken to be adequate.

At −10 deg F and neighbouring temperatures two factors intervene to make starting especially difficult. First, a greatly increased resistance to rotation prevails in all the moving parts of the engine and, secondly, there is a marked decrease in the sources of power to ensure starting.

On the first count, at very low temperatures even special winter oils become thick and near their "pour-point." This much increases drag between pistons and cylinder walls, big ends and crankshaft, crankshaft and bearings, and also between clutch and gearbox parts (Fig. 1). The engine is then hard to turn over, and difficult to keep running even when it fires.

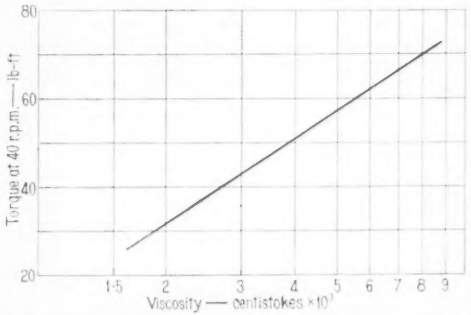

Fig. 1a. The increase in effort required to crank a 3½-litre engine as oil viscosity rises. The 80 to 120 r.p.m. required for a 1-litre engine will need still more torque

At the same time the electrical side loses efficiency due to drop in battery power. Furthermore, the evaporation rate of the fuel decreases as the temperature falls.

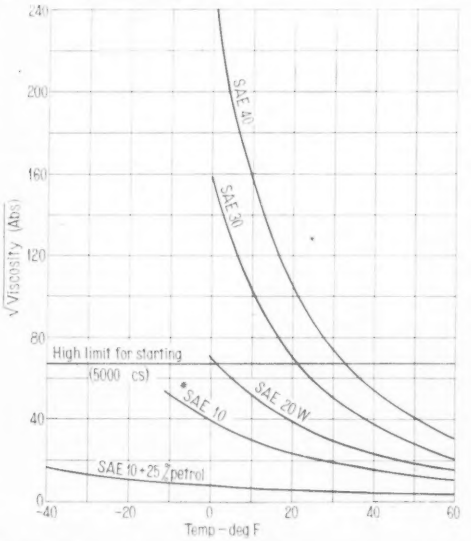

Fig. 1b. Engine cranking torque variation at constant speed with undoped oils (torque being proportional to viscosity on Barrington's formula). Viscosity may vary between oils of the same S.A.E. number. A 10W oil may reach 5,000 cs at −10 deg F

Drive-away from cold. Actually, once the engine is running, two different requirements have to be met. The driver may wish to move off immediately, or may decide to leave the vehicle stationary to allow the heater to warm the interior of the car, that is, warming-up the engine without touching the accelerator or even the "choke" control. In either case, rapid induction manifold warming is required to avoid liquid petrol passing through the cylinders into the crankcase and diluting the oil. This rapid warming will also reduce overall petrol consumption; to a very marked degree when the car is being used only on short runs. It ensures, moreover, that the vehicle runs smoothly and pleasantly at the earliest possible moment after starting.

Care must be taken with the carburettor design to meet the varying demands of the engine under these different circumstances. Too much richness at any given point may cause stalling, as well as being mechanically bad for the engine. An over-lean mixture at any point will also cause stalling or spitting-back.

Universality of specification. In fulfilling the above conditions, care must be taken that the vehicle will also start and warm up efficiently in the tropics. This implies the use of some form of automatic adjustment to weaken or richen the starting mixture according to the ambient temperature, whether the main operation of the carburettor starting device be automatic or hand-operated.

The special difficulties

1. *Cold Starting with Small Engines.* The cold starting problem for small engines differs to an important degree from the problem with larger engines. An arbitrary division of the two classes can be taken to be at about 2 litres. It is from the difference in capacity between the smaller British engines and the large American ones, that the main difference in the cold starting problem arises. Naturally, the engine will only start and run when the motive

*Engineer, Solex Limited.

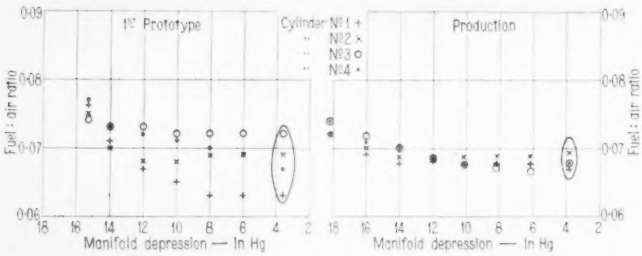

Fig. 2. Mixture distributions to individual cylinders, at low r.p.m., part throttle conditions. Air temperature 68 to 77 deg F. Left : manifold from first design. Right: manifold eventually adopted. Road performance confirmed this improvement

power developed is greater than the frictional resistance of the engine.

The first explosions at low temperatures are not strong, as the mixture in each cylinder is not an optimum one. The various surfaces are very cold, "filling" of the cylinders is less than desirable, revolutions are low, and the low piston speed does not give much help in raising the mixture temperature by compression, or in vaporizing the liquid petrol in the manifold by depression. The actual start will not be easy, for when the engine fires for the first time it may not develop enough power to overcome internal friction sufficiently and force the next piston to compress and give a second explosion. Before a subsequent explosion can occur, the crankshaft of a four-cylinder engine must rotate through a larger arc than is the case with a six- or eight-cylinder engine.

Friction is thus a much more important consideration with small engines. As an example, one may take a 500 cm^3 engine and a 4,500 cm^3 engine (both, say, four-cylinder units to simplify the illustration). Although the cubic capacity of the smaller engine is one ninth of the larger, the frictional surface area is only reduced to about one quarter. In fact, the motive power, which depends on the swept volume of the engine, varies according to the cube of the linear dimensions, while the frictional surface area varies according to the square. Thus the 500 cm^3 engine has a frictional resistance which in proportion to its cylinder capacity is about twice as large as a 4,500 cm^3 engine. It is largely for this reason that the small engine presents a special problem in very cold climates.

It may be noted in passing that, in addition to friction in the engine itself, there is considerable subsidiary friction, notably in the gearbox. Among the cold-starting instructions given in the manufacturer's handbook, should be a strong recommendation to declutch before attempting to start. Because of all these sources of friction a greater degree of care has to be given to a number of points in the vehicle specification; not only more care than we had to take before the war, but more care even, than the manufacturer of a large-capacity engine has to take.

2. *Carburettor.* Generally speaking a cranking speed of between 40 and 100 r.p.m. is all that can be expected at very low temperatures. The depression created by the engine in the induction manifold and consequently in the carburettor is a function of the cranking speed and therefore is likely to be rather low. Since the engine needs a much higher quantity of petrol vapour, the only way to create this is for the carburettor to supply an initial mixture rich enough to thoroughly wet the manifold walls with liquid petrol. From this liquid petrol, the more when it is a winter fuel, will "rise" a certain amount of vapour, partly from the volatility of the petrol itself and partly from the effects of the light depression created by the rotation of the engine.

This amount of vapour must be in sufficient strength in the air being admitted to form an explosive mixture. Unless factors other than the influence of the carburettor proper are suitable, the only way of ensuring that the mixture actually passing into the cylinders is rich enough will be to reduce the air supply to near, or even below, the safety line. If, however, the other factors are covered, the safest start can be obtained by providing a good deal more air to increase the power of the first explosions by "filling" the cylinders more adequately (Fig. 6).

As a small engine is liable to fire without picking up, a lack of air will be felt very quickly; it will, in fact, impede pick-up and thus strongly tend to drown the engine. This is one of the reasons for using proportionately more air than on large engines. Engines which are of larger capacity and have more cylinders are less trouble, as the very first explosion produces sufficient power to compress the mixture in the succeeding cylinder, that is, to pick-up and run. The same risk of drowning is not run by using a carburettor starting device which provides very little air and a lot of petrol. In any event, as soon as the engine fires the initial starting mixture must at once be weakened, as manifold depression from then on will rise considerably. This weakening is produced automatically on most types of carburettor.

3. *The influence of the Induction Manifold.* The first consideration is that, at low velocities, there should be an even distribution of vaporized, and liquid, petrol to the cylinders (Fig. 2). Next, there should be as large a surface area for evaporation of liquid petrol as possible. Usually the best way is to have a flat "floor" to the manifold. Also, the design should be such that the minimum quantity of liquid petrol should enter the cylinders. This is essential not only to prevent undue dilution of the sump oil and to avoid a high cylinder bore wear rate, but also to avoid wetting the plugs.

A further point is the avoidance of the possibility of an excessive amount of liquid petrol building up in the manifold. Provision should be made for a collecting point; this "reservoir" being subsequently emptied by the same means as is used to deal with drive-away, outlined later. The possibility of the contents of this reservoir being picked up in gouts and whisked into the engine as the latter fires must, of course, be avoided. This point remembered, usually the best manifold design is of the "ramshorn" type, and at least partly of a cross-section having a flat "floor" and an arched "roof."

Lastly, one of the most awkward points is the influence of the volume of the manifold. If the manifold is of small capacity, the depression communicated to it each time an inlet valve opens can be at a very useful value; if the manifold capacity is large the depression available to help vaporize the petrol will be slight. It is unfortunate that so much importance must, of necessity, be given to maximum b.h.p. figures with small engines. The high rate of r.p.m. necessary to achieve maximum possible power, imposes a manifold with too large a volume, due to the large cross-sectional area, to provide anything like the best conditions for very cold starting.

4. *Cylinder Head Design.* Even good manifold design cannot prevent a

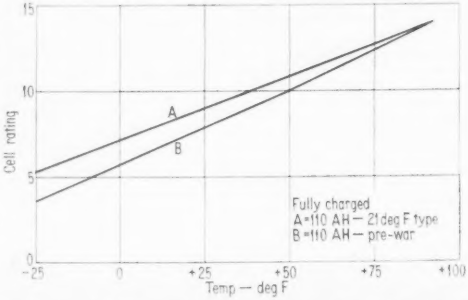

Fig. 3. The effect of temperature on cell-rating of two fully charged batteries. "Cell-rating" is the current in hundreds of amps discharge, required to drop the battery to half-voltage in 10 sec

certain amount of liquid petrol entering the cylinders, but cylinder head design should be such that the sparking plug is protected from this liquid and remains dry under such conditions. This is normally not difficult with overhead valves and it is well known that this type of engine is more easy to start than that with side valves.

5. *Ignition.* Ignition plays a very considerable part, and the user of a vehicle must be relied upon to keep the plugs clean. When moisture is deposited on a sooty plug it will very rapidly prevent a spark and, in practice, this can be one of the biggest troubles.

The main problem for the makers of the vehicle is to avoid coil-robbing, as low temperatures reduce battery efficiency (Fig. 3), and the starting motor may drain power sufficiently to prevent a spark. Usually it is considered that to produce a spark at half the normal tension is a satisfactory acceptance line on this point. Time of discharge of the spark must be long enough to ignite partially vaporized petrol of lowered evaporative powers.

6. *Starting Motor.* Although a cranking speed of over 100 r.p.m. would really be best on very small engines, this often implies a battery of far too large dimensions. The alternative is to use the starting motor to ensure that the engine continues to run after it first fires.

The problem is that with an inertia-engagement type of starting motor and 6-volt electrical equipment, as used in America and on the Continent, the frictional resistance of a small engine usually stalls the engine after the first explosion; which explosion, however, has caused the pinion of the starting motor to throw out. In practice, the further applications of the starting motor required will most likely drown the engine, manifold and plugs, before it can possibly pick-up and run. Two solutions are used to overcome this problem, and it is a basic one, in view of the fact that even when the engine fires, it is still only the starting motor which can be used to free the engine and give it most of the necessary momentum to run of its own accord.

In the United Kingdom we have adopted the 12-volt system which, in fact, has a large number of subsidiary advantages. Primarily, however, a 12 - volt starting motor will develop something approaching twice the b.h.p. of a 6-volt motor of the same dimensions (Fig. 4). This gives the armature an increased acceleration which in practice keeps the pinion in engagement with the flywheel for an appreciably longer time before throwing out. The alternative solution, retaining 6-volt units, is to use a "positive engagement" type of starting motor. Here the pinion is held in mesh physically as long as the starter knob is being pulled, or pushed, by the driver, up to a certain r.p.m. when the starting motor either throws-out or "over-runs."

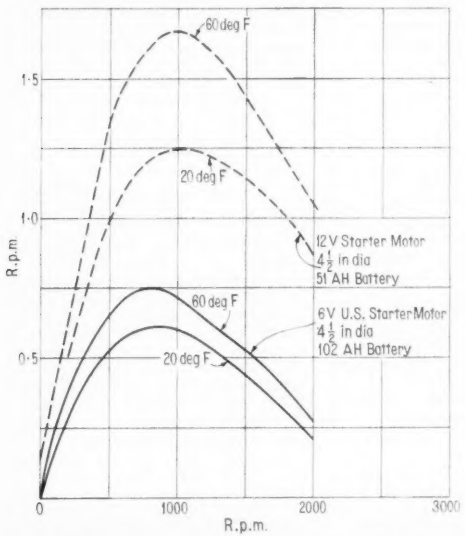

Fig. 4. Comparison of 12-volt and 6-volt starter motors of the same size and weight

While the safety margin for starting of the 6-volt, positive - engagement type is probably the larger, the 12-volt solution appears perfectly adequate, and the latter scores on almost every other point. The only argument usually made against it is the weight and cost of the batteries, but on a small car this is offset by saving in weight, cost and complication of the starting motor and dynamo. On a large car the weights are nearly the same. For other reasons there is a move in thought in the U.S. towards 12-volt systems.

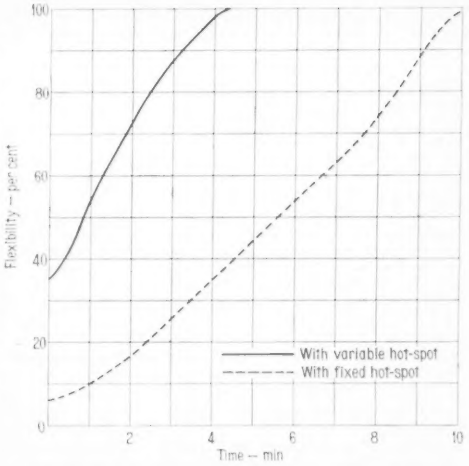

Fig. 5. Handling of the same car with and without thermostat hot-spot. After starting, the engine is presumed to run for 30 sec before attempting the drive-away, being gently accelerated once, prior to letting in the clutch

The combination of a 6-volt battery and an inertia-type starting motor, though satisfactory on large engines, will certainly not provide a proper solution to starting very small engines below about 10 deg F. With either of the two solutions outlined, an improved setting of the carburettor can be used, whereby the larger quantity of air will help to avoid plug-wetting.

Drive-away and warming-up

After the engine has begun to run, steps must be taken to ensure that it reaches normal temperature as soon as possible. It is necessary to heat the induction manifold, and also to warm the water reasonably quickly, and with it the cylinder head. The first is effected by using the exhaust gases, and the second by a thermostatically controlled valve to blank off the radiator water until a certain water temperature is attained in the cylinder block. It will, in the meantime, be necessary to provide the carburettor characteristics to suit the desiderata outlined earlier.

Finally, provision must be made to overcome the continuing frictional resistance of the oil, which will not reach its proper temperature until some time after the manifold should be warm. A mixture producing more power than the ordinary idling supply, one which also contains more air, is needed. This may or may not entail "fast idling," as the necessary mixture richening, by the extra power it produces, may be set merely to maintain normal idling r.p.m. until the oil warms thoroughly and engine speed consequently rises. When cold, the "oil-drag" may represent an increase of resistance of 30 to 40 per cent over idling resistance when warm, and

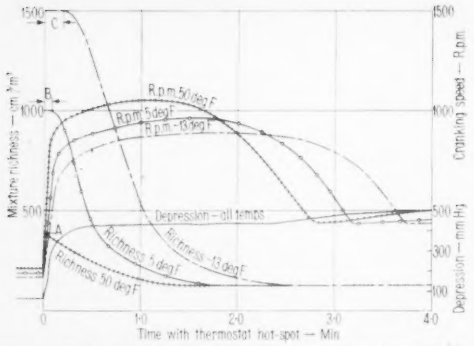

Fig. 6. Ideal carburettor characteristics, manually or automatically obtained, for start and warm-up

unless the engine is allowed to breathe more freely, it will stall.

1. *Carburettor requirements.* Whether the carburettor starting device be manually operated or automatic, the richness and volume of the mixture supplied, with the throttle closed, must progressively decrease as the manifold warms up. This is to give an engine speed, without load, and decreasing as the starting device is progressively dispensed with, from a rather high figure to normal idling speed. The aim is to avoid: damaging moving parts, dilution, manifold cooling, high consumption and piling up, or alternatively, starving (Fig. 6).

With the throttle opened for driving away, an enrichened mixture may have to be supplied automatically. The cold manifold walls provoke marked condensation of the atomized petrol and the final mixture reaching the cylinders may, unles enrichened, be too weak. The degree of enrichment should be such that the mixture reaching the cylinders is of the same strength as when the engine is warm. If it is enrichened much beyond this, excessive dilution will occur.

There are considerable differences between engines as to the degree of enrichment required. At about – 5 deg F, 20 per cent enrichment is an average figure on all power and part-throttle curves. If the engine is one which is on the rich side at full throttle when warm, slightly less than 20 per cent will be required. Indeed, running of some engines is noticeably affected by over-enrichment. Obviously, the degree of enrichment on drive-away must be progressively reduced as the manifold warms up.

2. *Induction manifold requirements.* The use of exhaust gases to heat the induction manifold is not of itself sufficient to provide the answer to the problem of warming up. If the heat applied is only sufficient to maintain eventually correct running temperature, then warming up will be far too slow. On the other hand, if enough heat is applied to a plain hot-spot to warm up sufficiently quickly, normal running temperature of the manifold will be too hot.

It may be considered that a thermostatically controlled valve in conjunction with the hot-spot is normally obligatory, satisfying as it does the two conditions, as well as rapidly evaporating the liquid petrol present in the manifold "collecting point" immediately after starting. The difference in the use of the car in very cold conditions with a thermostat hot-spot fitted is remarkable (Fig. 5).

The alternative system, particularly with engine designs where induction and exhaust are on opposite sides of the engine, is water heating of the manifold. With this arrangement it is, of course, absolutely necessary to provide the thermostatically controlled valve to cut off the radiator water until the water around the cylinder block has reached a set temperature. However, unless the engine characteristics include excellent turbulence and extremely good distribution through the induction manifold, it should be considered that this method of manifold heating is an effective bar to the proper solution of the cold drive-away problem, at least on small engines. In any event, not only is this form of warmth very slow in making itself felt, but it may not rise to a temperature at which ordinary fuels will be nearly fully vaporized. It is more likely, on starting in very cold weather, that the water-heated manifold rather condenses than vaporizes the mixture.

Furthermore, dilution is a very important reason for specifying the thermostat hot-spot, in order that the carburettor starting device shall be put out of action at the earliest possible moment. It is during the period of use of the choke after the engine has started that the major percentage of dilution occurs. On continual short runs in cold weather, dilution can very quickly be very damaging to the engine and, in any circumstances, the life of the engine will be materially affected.

3. *Other factors.* Numerous other elements play their part and, again particularly with small engines, they can be important parts. Unavoidably, small car owners need to give closer attention to certain details than do the drivers of big cars, even when the best results have been obtained in setting the engine.

The fact that most batteries will, if allowed to fall to half-charged condition, freeze at say – 13 deg F, helps to keep owners aware of the need to check this. In addition, plugs and plug gaps, ignition timing, tappets and valves, and if of a type requiring it, the carburettor starting device, will always have a bigger influence on the performance of a small engine.

General considerations

The means of meeting all the foregoing conditions are mostly satisfactorily settled. Sometimes they are, however, the subject of involved, empirical work due to the characteristics of the particular engine. This is usually due to disagreement with one or more of the solutions suggested earlier and summarized in the concluding section.

Sometimes alternative methods of providing these solutions are the subject of involved technical discussion. The most interesting discussion of this nature is that of the rival protagonists of the starter-type carburettor and the strangler-type carburettor, concerning the degree of flexibility of each towards giving an ideal performance at every different stage of cold-starting and warming-up (Fig. 6).

Following the original "self-starter" carburettor, later largely replaced by the "bi-starter" and more recently by the "zero starter," available with "fast idle" and enriched drive-away, we have finally the "progressive starter" with all and more of these latest refinements.

The original simple strangler has been largely replaced by the spring loaded "off-set" strangler and a number of other developments have been made from time to time to improve the device still further, and to suit the same wide variety of conditions as led to "starter" development.

Both starter and strangler are used in large quantities on small engines, the starter type having a predominance. In the author's opinion it is more nearly possible to meet all conditions with the starter type. Those conditions include a point not hitherto mentioned; that of misuse by the private owner. Briefly, as these discussions are necessarily involved, there are three main points as well as other lesser ones which seem to give the starter type an advantage.

First, the starter has little risk of going out of adjustment which, when the difference between a start and a failure is so small, becomes an important factor in service, if not so readily apparent on the prototype test bench.

Secondly, the setting of the starting mixture is made independently of the setting of the main carburettor circuit. This allows both petrol and air to be supplied in the quantities required, without need for consideration of the main setting for normal running. From this, for example, greater ease of limiting engine speed to a reasonable level during warming-up is one of the lesser advantages. It may be difficult on a small engine set for economy, to pass enough petrol through the main jet, with a strangler, to ensure starting.

Thirdly, the starting mixture is not so easily varied, at least adversely, by the driver's actions, which in these particular conditions may have an

unusually important influence.

There are two main criticisms of the starter unit. First, that prototype setting needs a little more care than with a strangler type, although this is felt to be justified by increased safety from drowning, and simpler use. Secondly, that the clear-cut difference in this case in the method of manipulation by the driver is adversely commented on by users firmly trained in American technique on large cars, as is the case in Canada.

However, correct manipulation of the starter is a very simple matter indeed; a very small part of accustoming oneself to a small car as a whole and comparable to getting used to the gear shift on a new car. The clear-cut difference in manipulation may indeed be an advantage in view of the difference in behaviour under conditions of extreme cold of a small engine. Starter manipulation is really only commented on when, for one reason or another and mostly other than faulty use of the dashboard knob, the vehicle does not start.

It is open to question whether the strangler is not just as liable to give difficulty. Although its operation should apparently be the same as for a strangler on a large engine, the difference in the requirements of a small engine often make this similarity of manipulation deceptive. In fact, starting difficulties, which existed in Canada over two successive winters just after the war, showed that further development of both devices was at that time necessary, and these improvements have since mostly been made.

The whole question, as stated earlier, is a very involved one, and hinges on which of the two techniques has the better inherent ability to meet the full problem, which includes the influence of so many other elements of the vehicle specification, and of the driver.

Conclusions

To have sufficiently satisfactory starting, warming-up and drive-away with small engines in very cold climates, a carburettor starting device should be used which gives proportionately more air than with a large engine, as well as a large quantity of petrol.

There will be little manifold depression for starting due to the need for a large cross-sectional area to extract the maximum b.h.p. from the engine; this allows the carburettor setting above but also underlines the need for a 12-volt system or a positive-engagement type starter motor.

One of these two systems is required to ensure that with the first explosions, enough power is supplied to allow the engine to run, since frictional resistance plays a much more important part on small engines than on large-capacity units.

After starting, the carburettor starting device must weaken the mixture at once and provide for progressive weakening as the manifold warms up. Fig. 6 shows the ideal requirements. It may be noted that an automatic device has difficulty in obtaining the characteristics marked B and C. Here, the periods are too long for manifold depression to be utilized, but too short for manifold temperature to be employed. On the other hand the mixture must be richened as the throttle is opened to drive away.

Induction manifold distribution at low velocity must be good, and manifold design should avoid excess liquid petrol entering the cylinders. Thermostatically controlled valves must be provided in the hot-spot arrangements and in the water circulation. Batteries must be of larger capacity in proportion to the engine capacity than is the case with large engines. The carburettor should incorporate an automatic device to lessen the difficulties of meeting tropical and sub-arctic use, by automatically varying the starting mixture strength according to ambient atmospheric temperature.

That a small-capacity engine calls for more care from the driver than does a relatively large engine cannot be avoided. The simple facts must be learned of how to make the best use of available controls, including declutching when starting from very cold, in order to avoid any difficulty. Regular attention must be given to battery, plugs, ignition timing, tappets and valves. The use of high V.I. thin oils of the best possible quality is also more important — "waxy," thick, or low V.I. oils must be avoided.

From the point of view of British car and accessory manufacturers, it is unfortunate that extremely cold conditions are not indigenous to enable a normal utilization of vehicles to be observed. This makes empirical confirmations of the correctness of many decisions difficult to obtain, and even interferes with cold room work.

Although a number of problems were soluble from a study of American practice, some cannot be approached this way since the significance of the differences in engine capacities intervenes. Perfection has not been attained, and it is no unusual sight in Winnipeg, for example, at "forty below" to see cars having large-capacity engines immobile through failure to start. The cause it not always ice in the fuel lines, and so long as such failures are the exception rather than the rule, the room for improvement will be slight. We have, indeed, just that attitude to punctures.

Grateful acknowledgment for assistance, and for data supplied for the preparation of this article, is made to Messrs. Standard Motor Co. Ltd., Joseph Lucas Ltd., Esso Petroleum Co. Ltd. and C.A.V. Ltd.

ERRATA

Gordon Aerolite

IN the article on Plastics Body Construction in the June issue, a table was published giving the physical properties of some laminated plastics and metals. We are informed by Aero Research Ltd. that the figures quoted for Gordon Aerolite, in the column headed E (lb/in$^2 \times 10^6$), and in the last column giving the specific properties, should have been respectively 7·0 and 4·89 instead of 2·0 and 1·4.

Vauxhall Engines

IN a table of valve data on page 329 of the September issue, the timing of the inlet valve closure was inadvertently quoted as 39 deg B.B.D.C. (before bottom dead centre). It should, of course, have been given as 39 deg A.B.D.C. (after bottom dead centre).

SHOW REVIEW NUMBER

The extra issue of the "Automobile Engineer" will be published as usual in connection with the London Motor Exhibition. It will constitute a critical review of the more interesting exhibits, including coachwork, and will have numerous illustrations of special features and design characteristics.

Publication of this issue will follow the ordinary November issue, which will appear on November 6th. The Show issue will be available on November 27th, and can be obtained by order from newsagents throughout the United Kingdom, price 3s. 6d. net. Readers are reminded that it is still necessary to make arrangements with the newsagent to make certain that a copy is secured.

Standard Diamond Tools

FOR general radius and form dressing of grinding wheels, a range of standard diamond chisel tools has been introduced by Engineering Diamonds Ltd., of 26, Warwick Row, Coventry. Lapped to high precision and finish, these tools are available in four standard angles, 60, 70, 80 and 90 deg inclusive, each with the option of nine edge forms. The range is thus comprehensive in application and reduces the need to manufacture against special specifications.

Standard holders are $\frac{5}{16}$, $\frac{3}{8}$, $\frac{7}{16}$ and $\frac{1}{2}$ in diameter, with lengths from 1 to 3 in. These dimensions, it is claimed, are suitable for the majority of forming operations. One tool, in a holder $\frac{5}{8}$ in diameter and 5 in long, is intended for hand form dressing and is standardized at 60 deg angle with a maximum edge facet of 0·002 in width. (2025)

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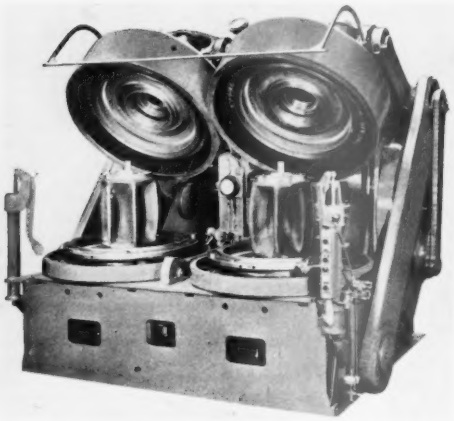

Fig. 1. Bag-O-Matic press ready for loading

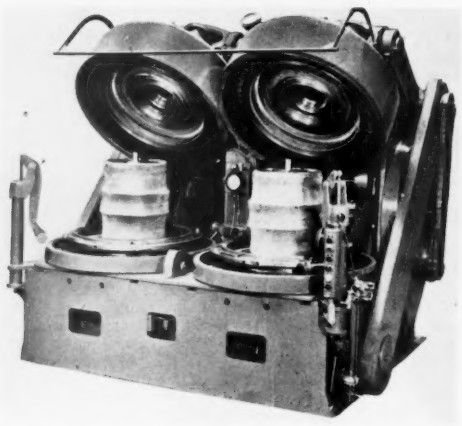

Fig. 2. The press with the green tyre in position

IT has long been recognized that there can be much wasted effort during the vulcanizing stage of tyre manufacture, that is, between the finishing of a green pneumatic tyre on a tyre building machine and the production of the finished article. Normally, after the green tyre has left the tyre building machine, it is handled five times before the finished tyre is produced. For a 6·70-15 tyre this entails a total handling weight of approximately 350 lb. Some few years ago, The Macneil Machine and Engineering Company, Akron, Ohio, U.S.A., undertook investigations with a view to shortening the curing time and reducing the handling. Eventually they evolved the Bag-O-Matic press. With this equipment the tyre is handled only twice, and in dealing with a 6·70-15 tyre the total weight movement is reduced to approximately 52 lb. Until recently the Bag-O-Matic presses were obtainable only from the U.S.A., but they are now being manufactured under licence by Francis Shaw and Co., Ltd., Manchester.

A Bag-O-Matic press ready for the reception of a green tyre is shown in Fig. 1. In this position the centre mechanism is extended vertically and the bags are collapsed to allow the green tyre to be placed in position as shown in Fig. 2. As soon as the tyre is in position, a completely automatic cycle is initiated by a push button. The press dome is lowered and the bag is expanded by steam to form the tyres into the mould. The press closes completely with a parallel action and remains closed until the cure is completed.

During the curing period, high pressure steam is automatically valved into the curing bags until the predetermined period has elapsed. At the end of this period, the dome and the top half mould opens automatically, to leave the tyre in the bottom mould. When the dome has reached its maximum elevation in the open position, the tyre is automatically stripped from the bottom half mould by the lower bead and bag clamping ring, and the centre mechanism begins to rise vertically. Before the bag is completely extracted from the tyre, the lifting arms automatically extend and hold it at working height. At this point the bag has been completely extended mechanically in a vertical direction and has been collapsed axially by the operation of a vacuum. This decreases the bag diameter so that it is smaller than the bead diameter of the finished tyre. Then, through a push button control, the bag is lowered and the tyres are left on the lifting arms as shown in Fig. 3. This completes the operation and the tyres are ready for removal. It is then only necessary to lower the lifting arms and the press is ready for reloading.

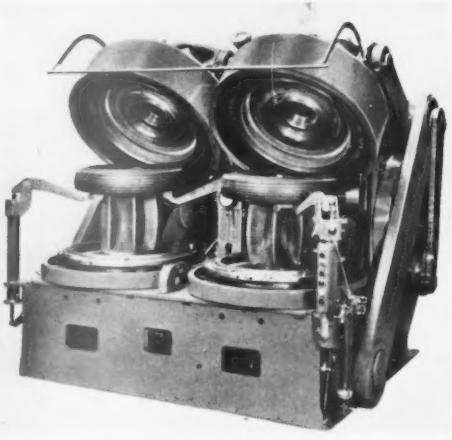

Fig. 3. Cured tyres ready for unloading

For passenger size tyres, such as the 6·70-15 range, the curing cycle time by Bag-O-Matic is between 18 and 23 minutes. Despite this short curing time, the ease of loading and reloading the press makes it possible for one operator to be responsible for a battery of presses. For example, one installation consists of 27 twin Bag-O-Matics, 54 mould positions, controlled by one operator.

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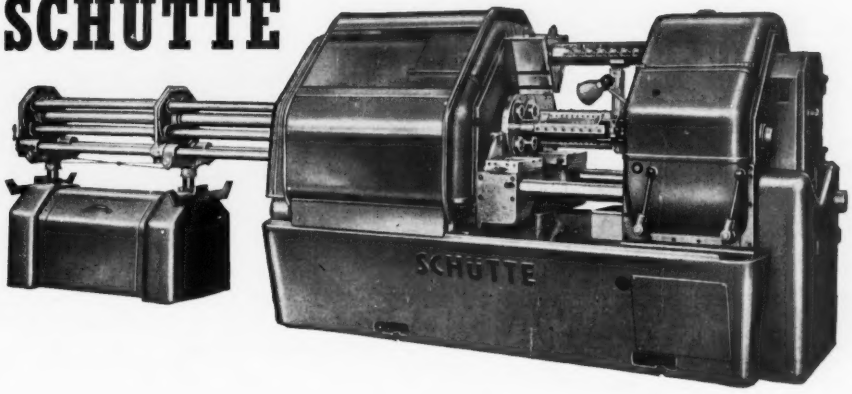

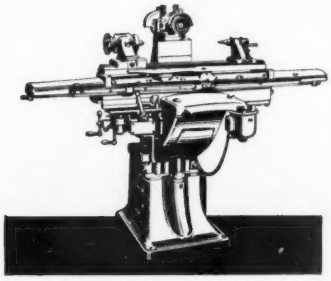

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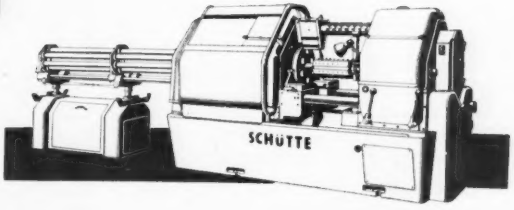

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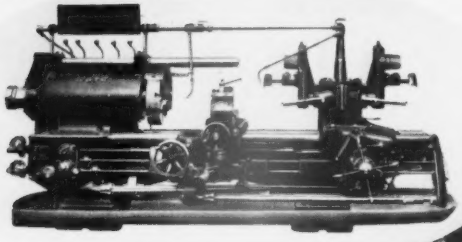

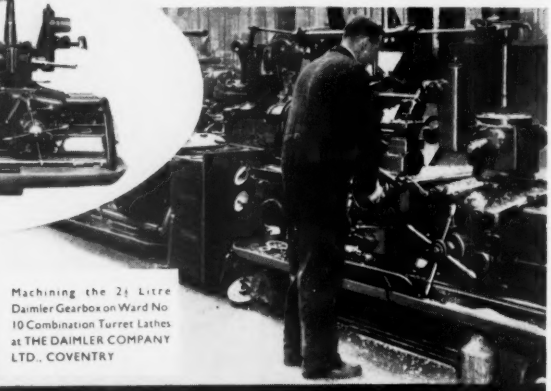

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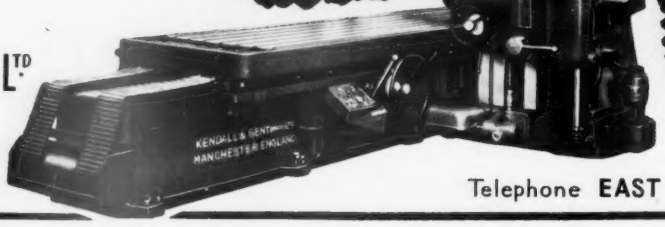

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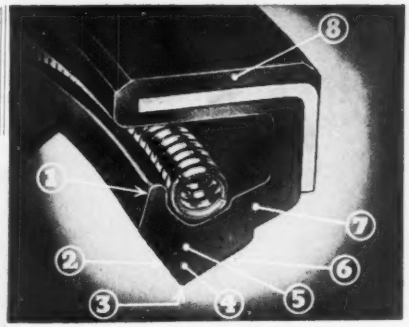

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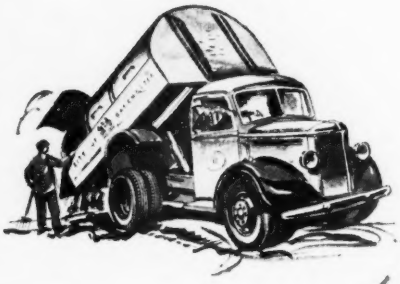

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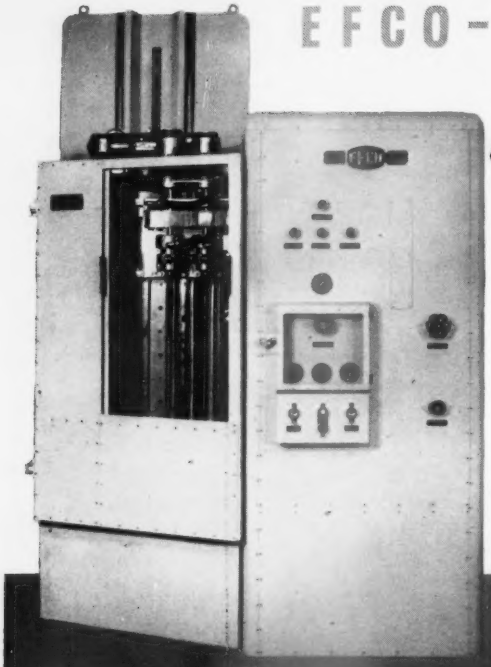

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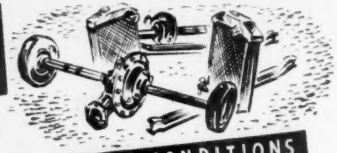

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"WEDCO" Woven Wire Belt Conveyor with side tables for Radiator Assembly. General view from drive end. Belt fitted with centre guide rollers.
Photo by courtesy of Vauxhall Motors Ltd.

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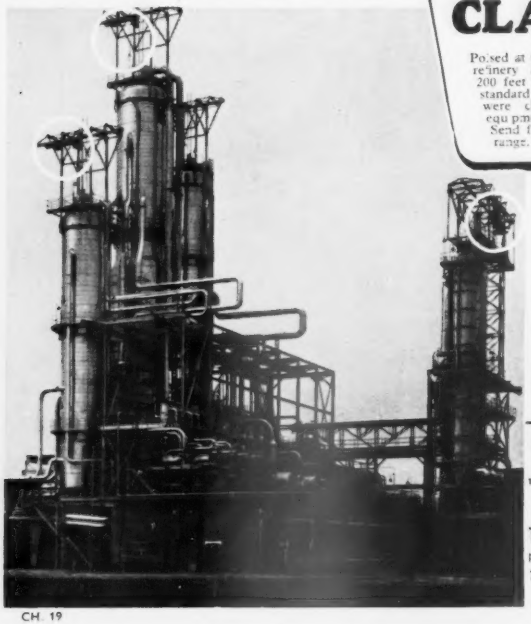

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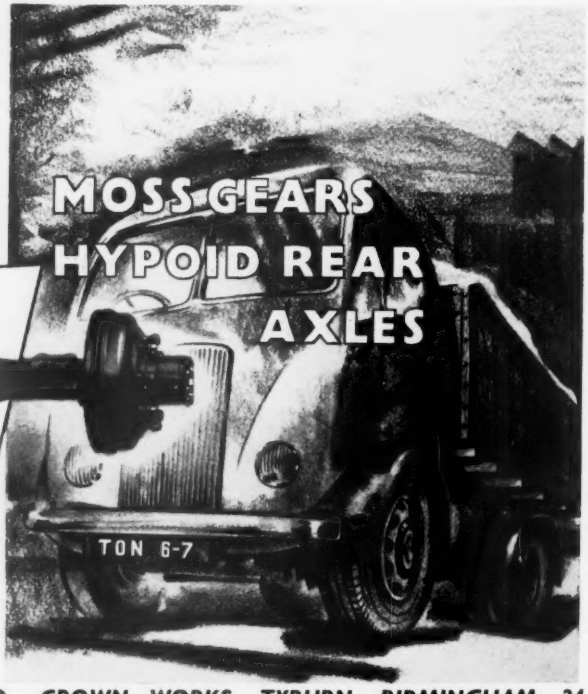

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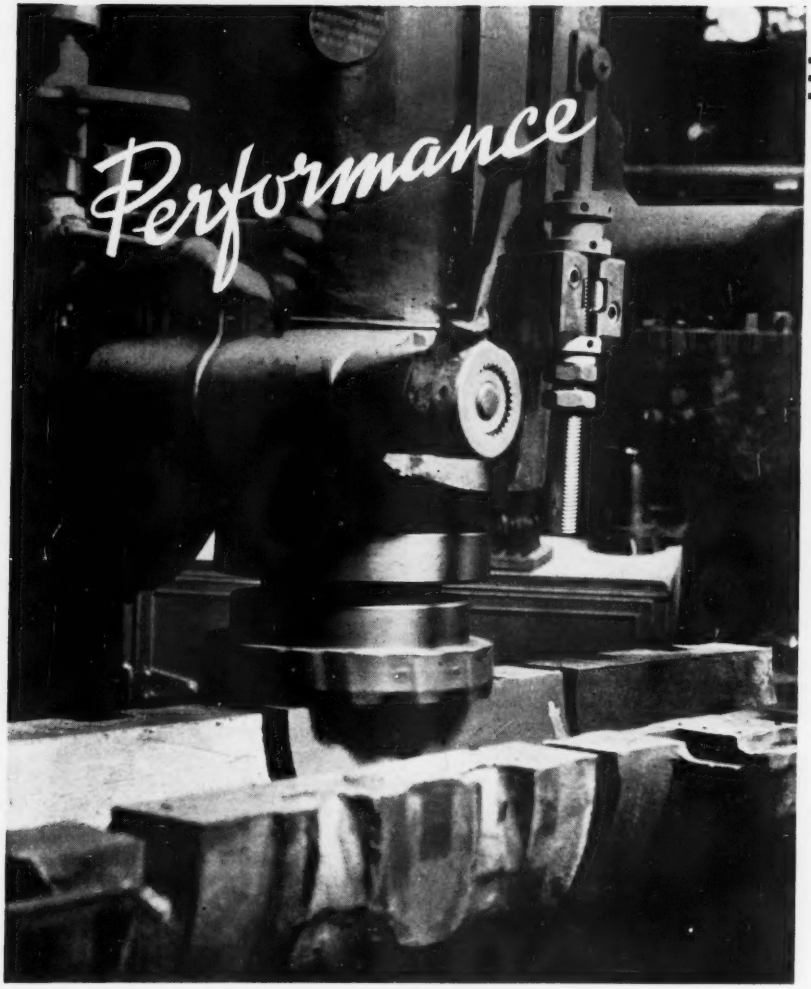

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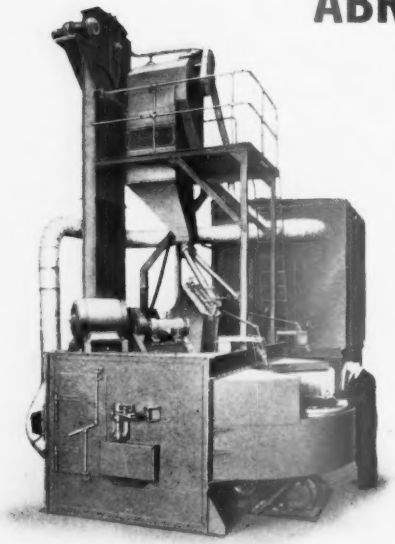

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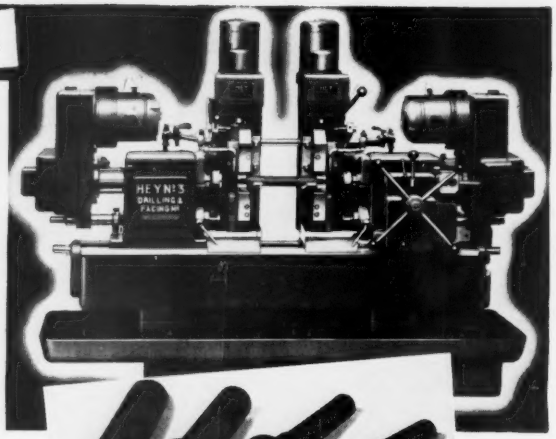

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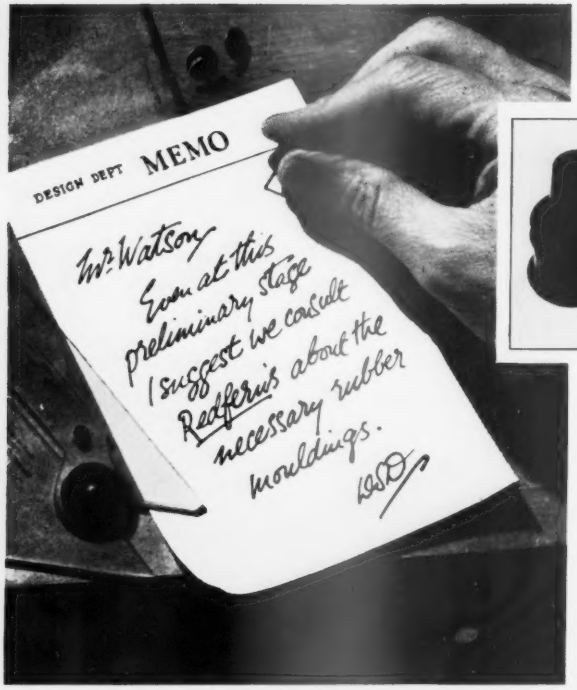
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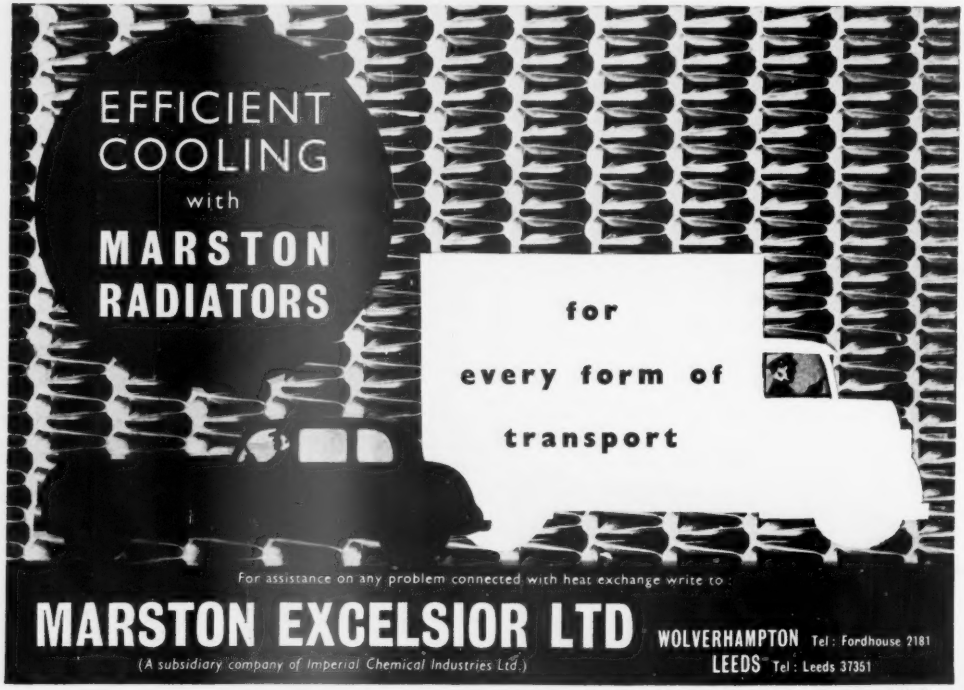

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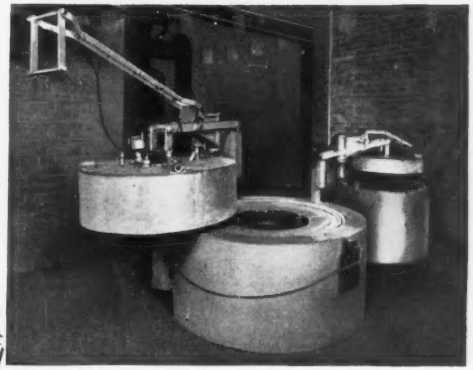
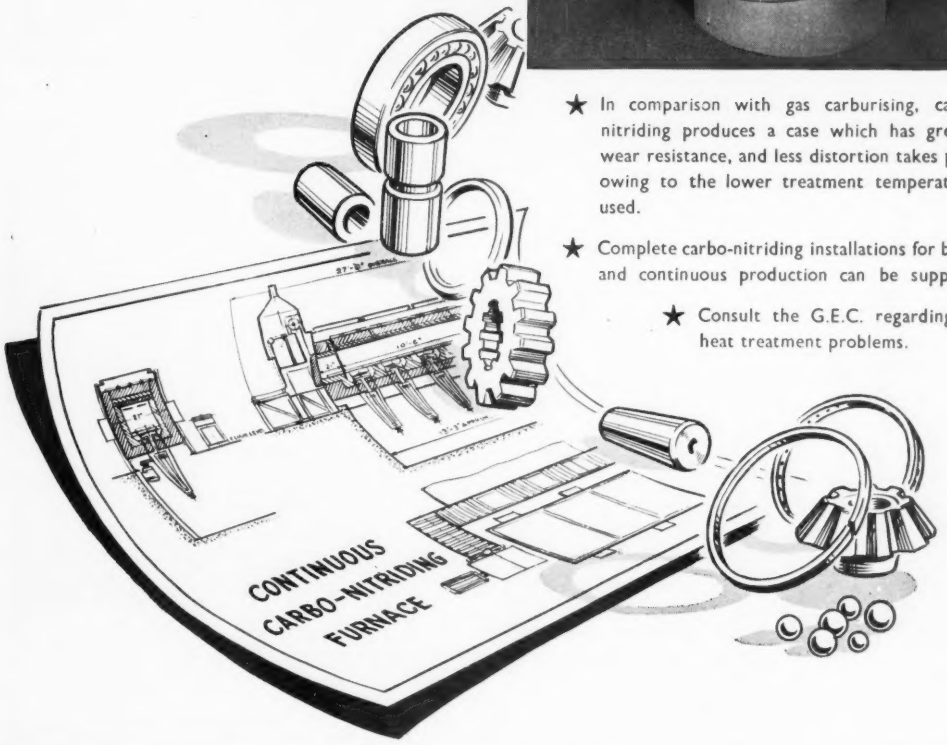
★ The process is cheaper than liquid cyaniding and the case composition as well as depth can be controlled. Cyanide waste is eliminated.



★ In comparison with gas carburising, carbo-nitriding produces a case which has greater wear resistance, and less distortion takes place owing to the lower treatment temperatures used.

★ Complete carbo-nitriding installations for batch and continuous production can be supplied.

★ Consult the G.E.C. regarding all heat treatment problems.



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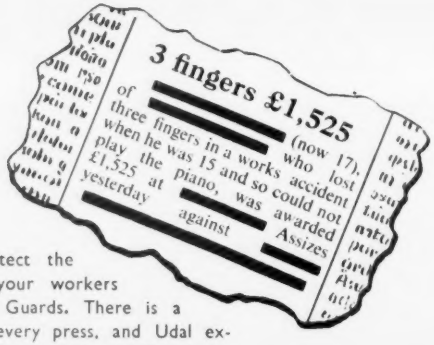

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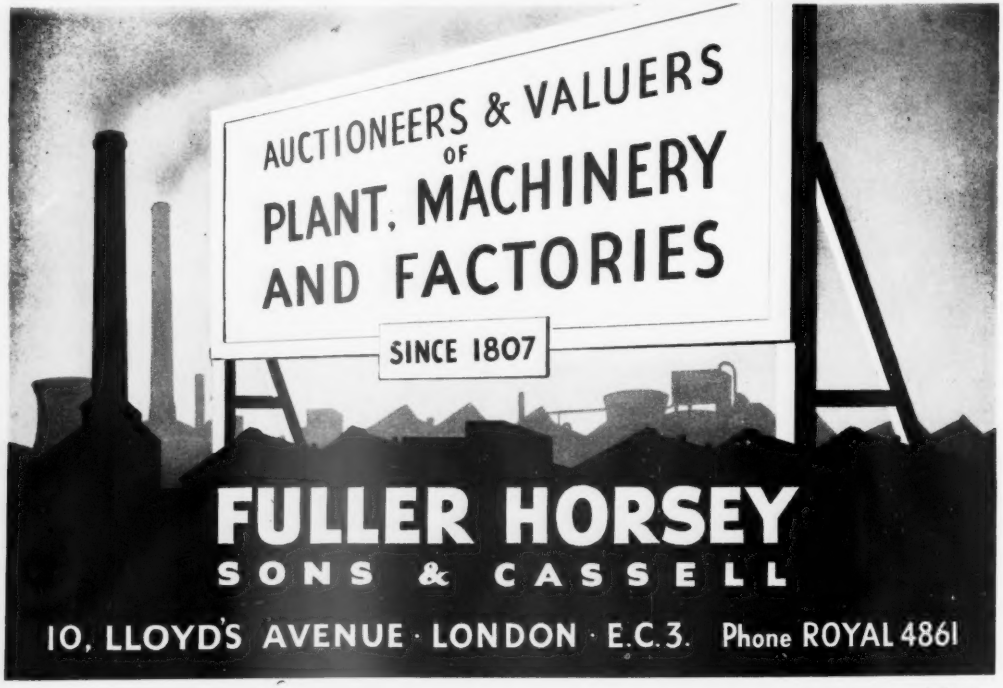
PRESS GUARDS



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stands for **all** plain bearings

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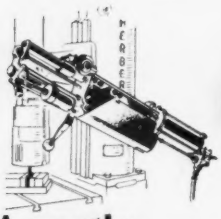

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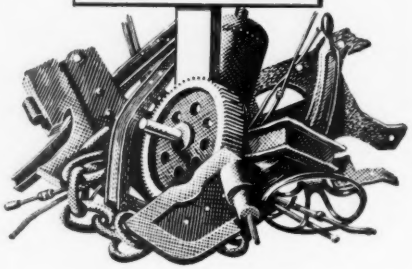
T.28

WHAT EVERY MANAGEMENT SHOULD KNOW ABOUT

scrap



What is scrap?

Obsolete plant and machines, old spares, redundant buildings—anything made from iron and steel for which you have no foreseeable use.

What good is it to me?

No good at all in its present state. But send it back to the steelworks and it can go to make the new steel that you and all steel-users need. Remember—a ton of scrap may make a ton of new steel.

Can't they make steel without scrap?

Yes—but not enough steel. Last year, 58% of our steel was made from scrap. The more scrap we use the more coal, coke and iron ore we save for other important purposes.

What should I do with my scrap?

See your local scrap merchant. He will be glad to help with dismantling and collection. If you have any trouble disposing of iron and steel scrap, write to the British Iron & Steel Federation for the address of your Joint District Scrap Committee.

What shall I get for it?

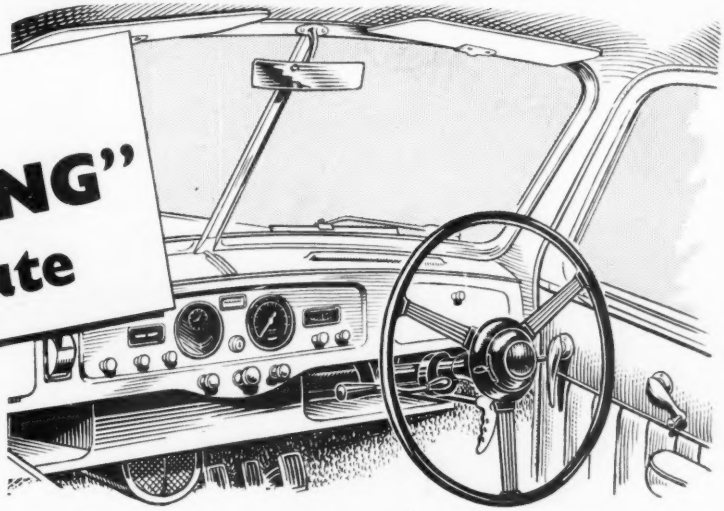
You will be paid a fair price for your scrap. MORE IMPORTANT—you will be doing yourself and all steel-using firms a great service by helping to increase steel production.

SPEED THE SCRAP

SPEED THE STEEL

Issued for the STEEL SCRAP DRIVE, by the British Iron and Steel Federation and the National Federation of Scrap Iron, Steel and Metal Merchants.

★ A "FITTING" tribute



★ LET US QUOTE YOU FOR THAT JOB

A fitting tribute indeed to the ingenuity of Plastomatic who would like to handle compression or injection mouldings for you. Our service is geared to the demands of the Automobile Industry. Within our production scope come every conceivable variety of parts and fittings. The list below covers some of these, but don't let your imagination stop there — ours doesn't! Why not consult us? The Plastomatic Service is at your instant command.

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of our Service:— (1) We will undertake the design for you, and guarantee your exclusive rights to it throughout the period of contract. (2) We produce high quality mouldings in large or small quantities — just name them!

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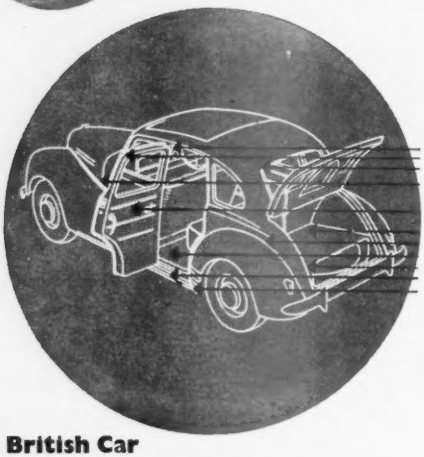
Don't just stick it.....

Bostik it!



ALL British Car Manufacturers use BOSTIK for one or more of these operations:

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The BOSTIK System of Adhesion includes Bostik Adhesives, Prestik, and Bostik Sealing Compounds developed to meet the motor industry's never-ending demand for faster, cheaper, and more efficient production methods.

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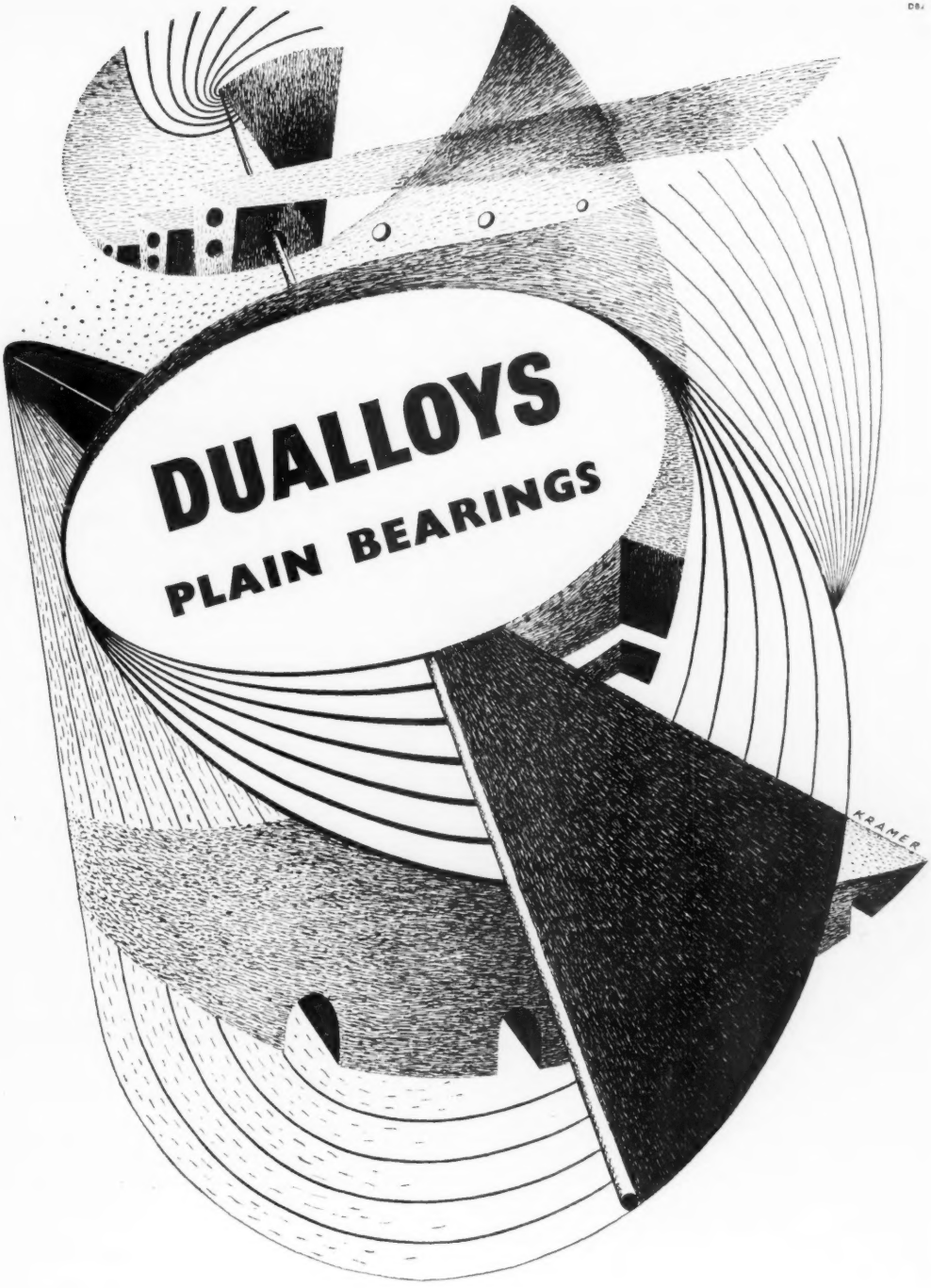

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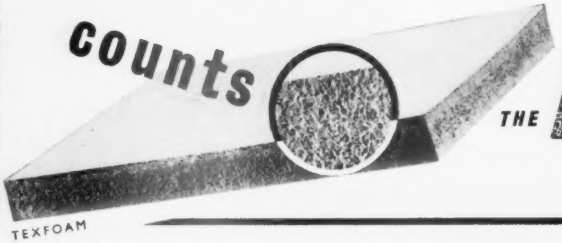

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Every move the passenger makes, every move the vehicle makes, and every move you make on seating design, affects travelling comfort. TEXFOAM and HAIRLOK have been put to the most exacting tests under the severest operating conditions and have proved that seating problems can be overcome at the minimum cost. We shall be glad to collaborate with you on any seating problem.

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P7

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Photo taken in the Austin Service Dept.

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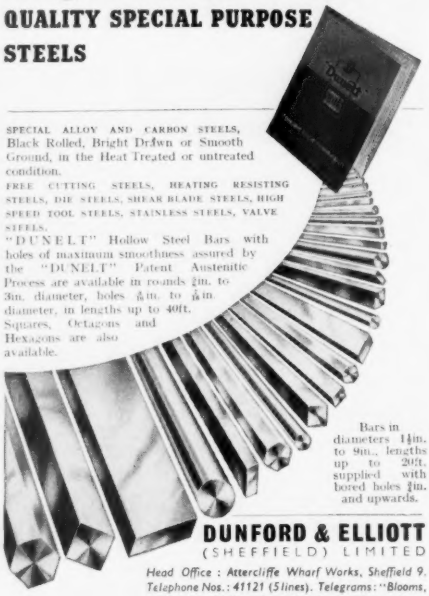

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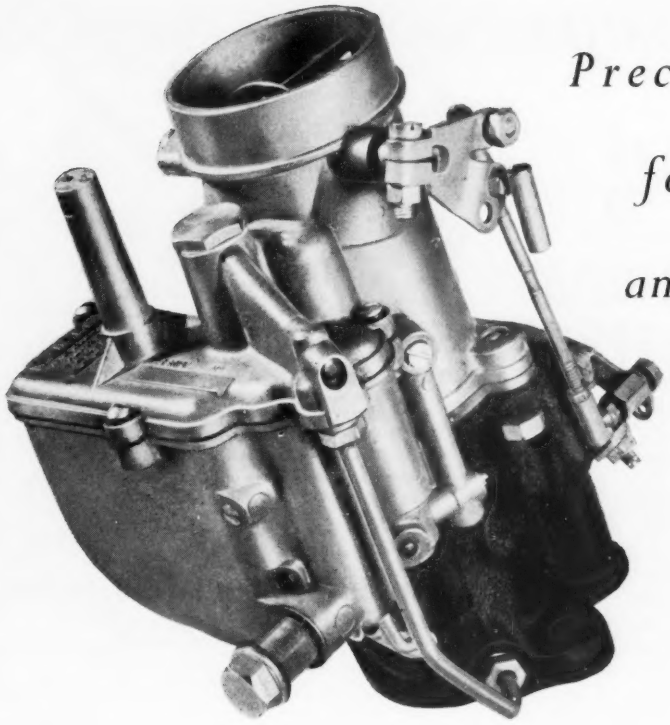

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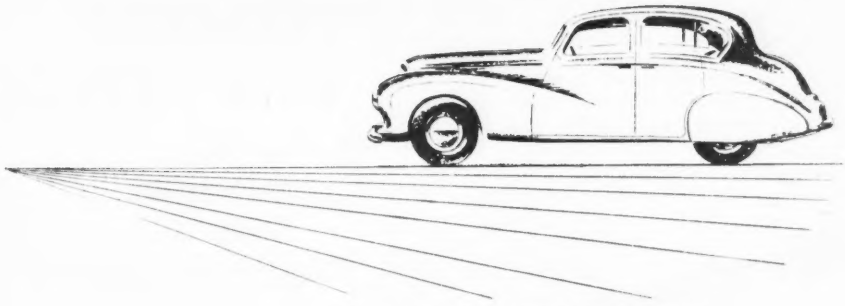

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Photograph shows I-C engine pistons treated with 'dag' colloidal graphite before running-in. (Reproduced by courtesy of The British Piston Ring Co. Ltd.)

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Illustration of graphited camshaft by courtesy of The Standard Motor Company Limited.

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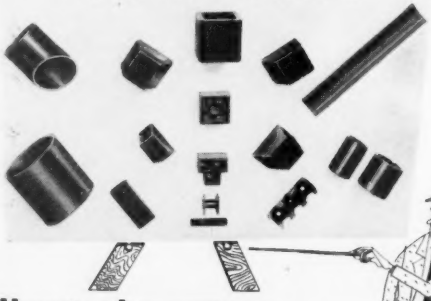

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(we can't guess what)

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(we're not clear how)

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(we wonder which)

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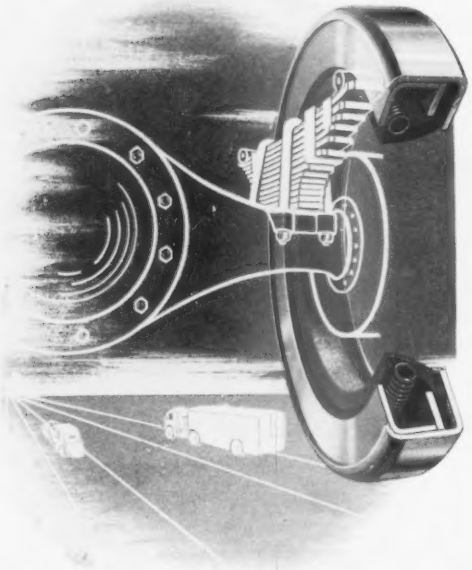
Insist on having new oil seals fitted when bearings are inspected or repacked with grease—the old ones are usually damaged on removal—and make sure that the seals refitted are Weston Oil Seals.



Whatever the machine, if it has shafts that turn, its performance will depend on the efficiency of oil seals. The maximum protection possible must be provided for bearings and WESTON OIL SEALS provide this protection. If you have any sealing problems, our technical staff will welcome a chance to solve them.

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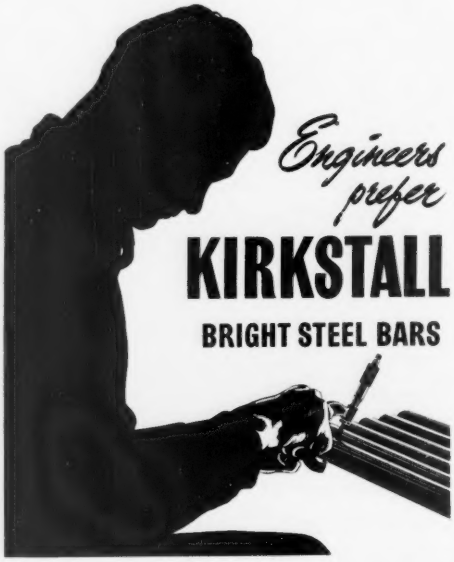

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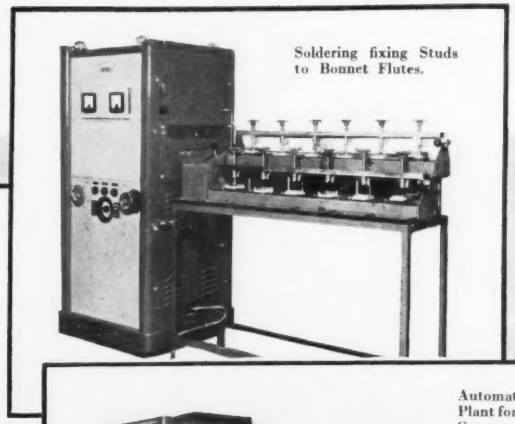
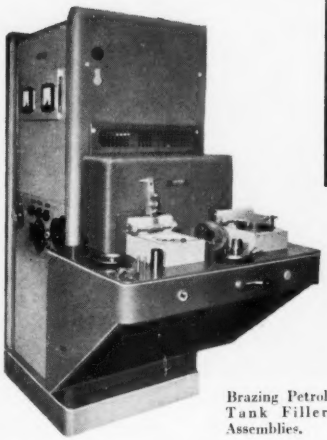
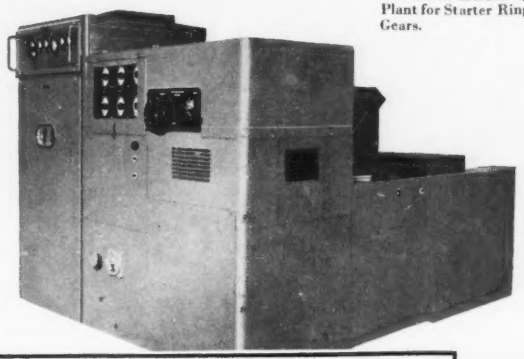


Soldering fixing Studs to Bonnet Flutes.



Brazing Petrol Tank Filler Assemblies.



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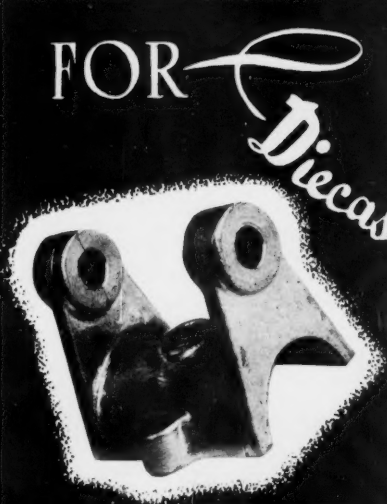

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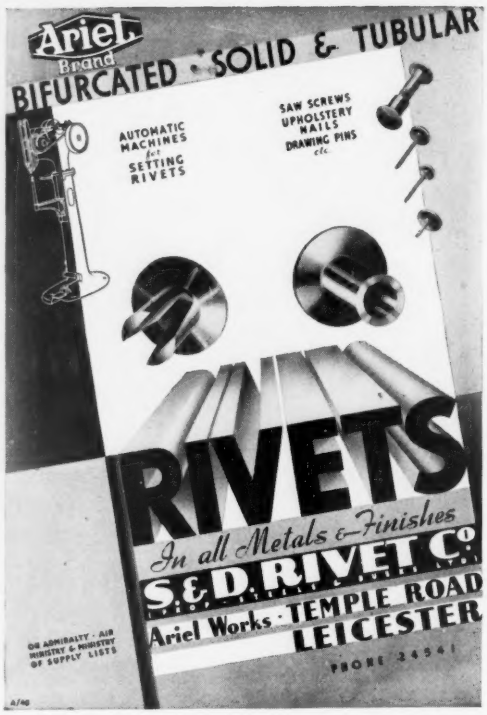
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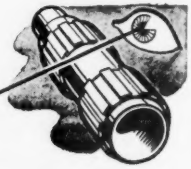

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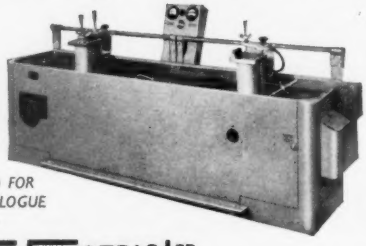

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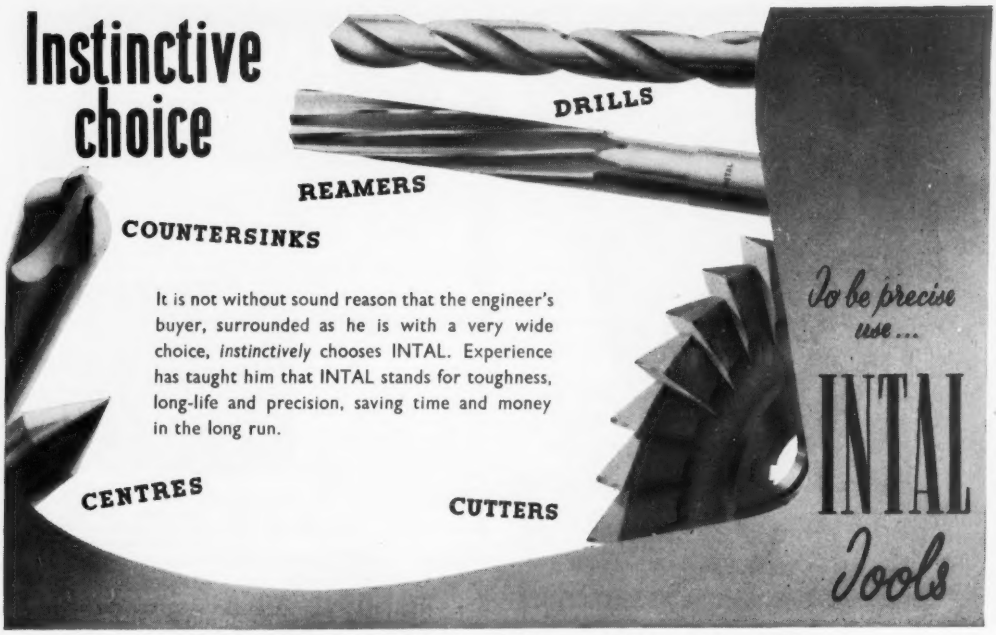

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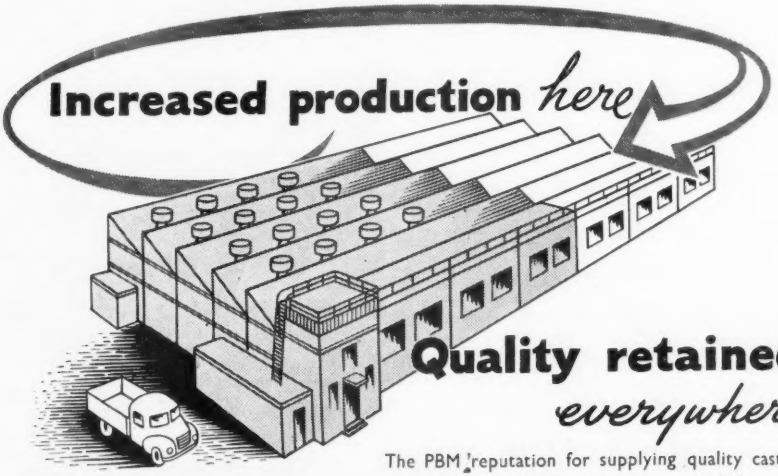

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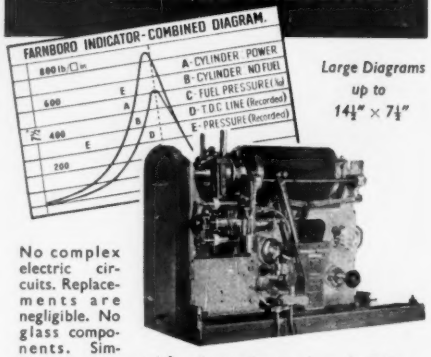

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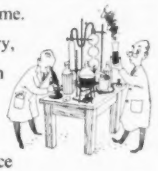

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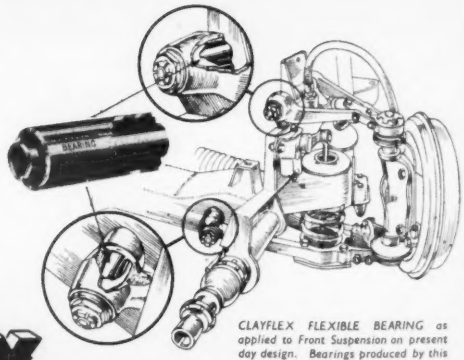

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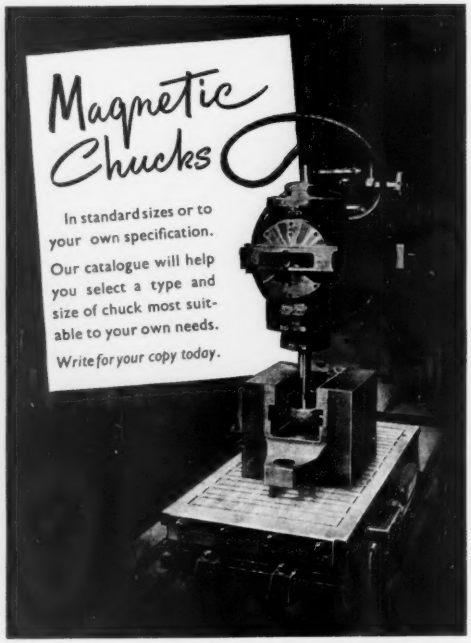
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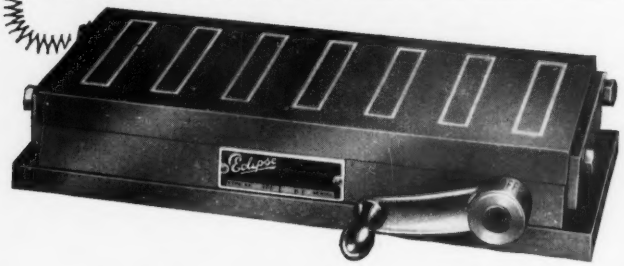

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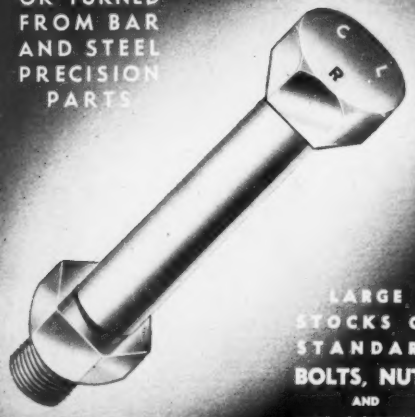

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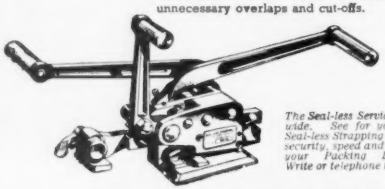

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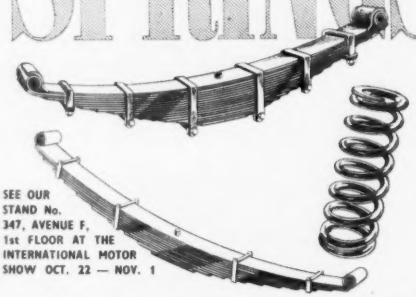

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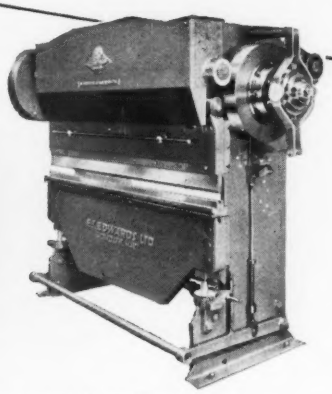

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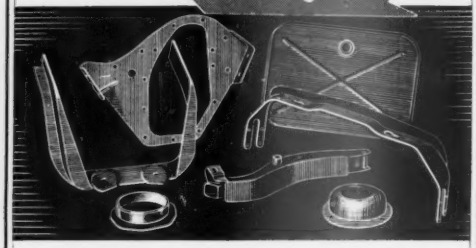

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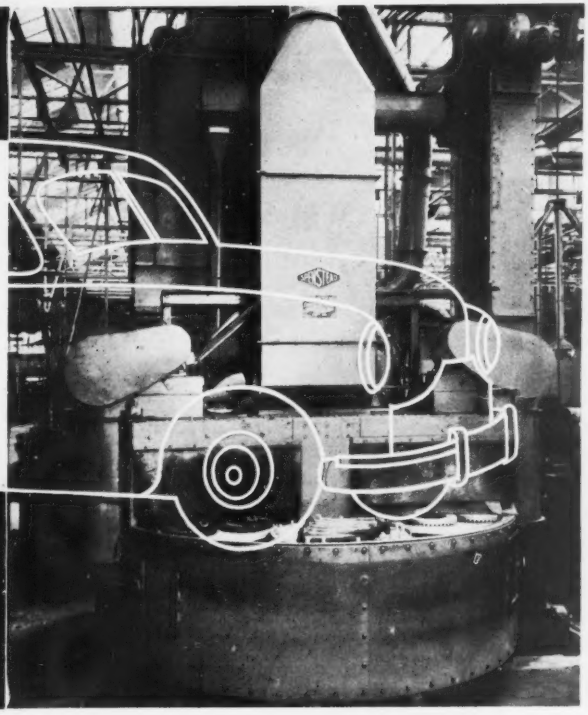

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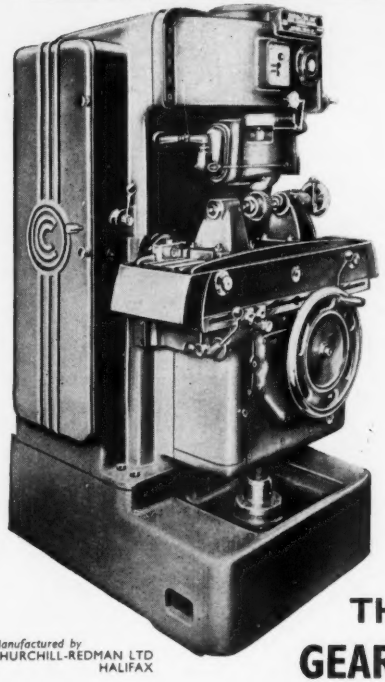

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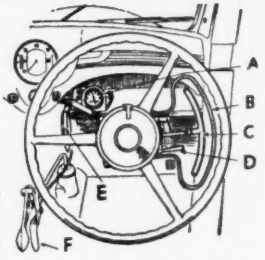

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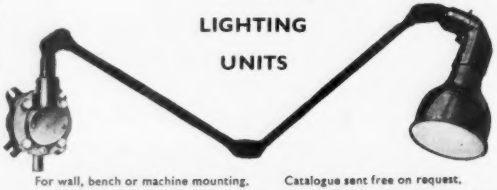

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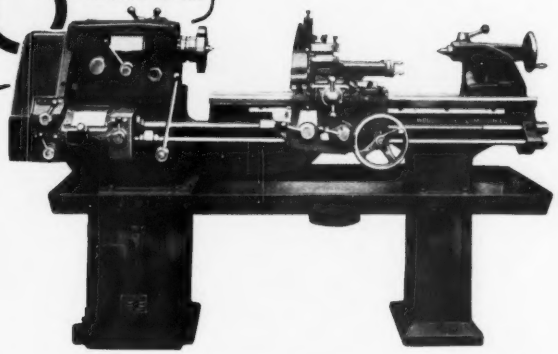

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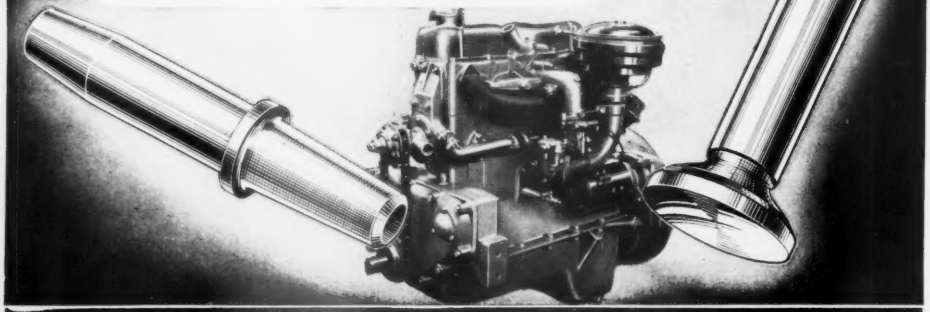

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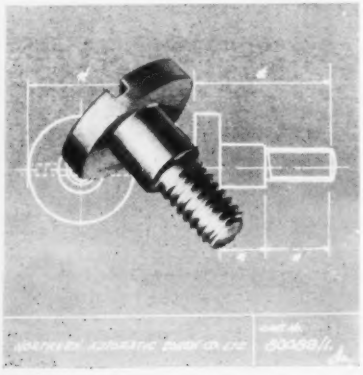

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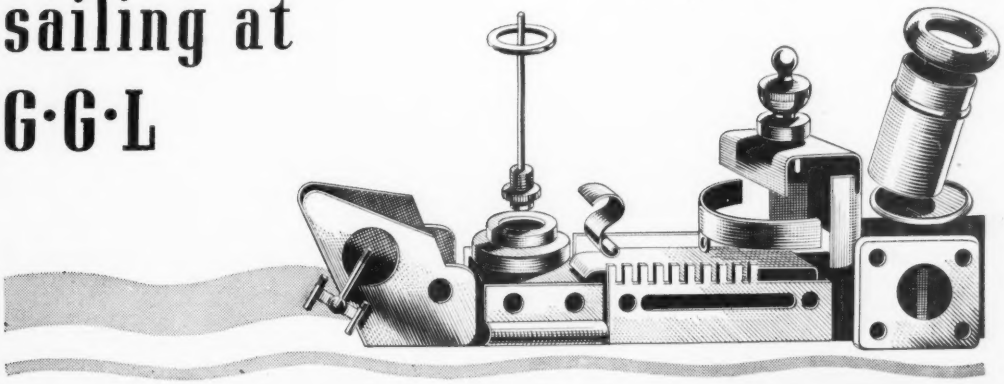

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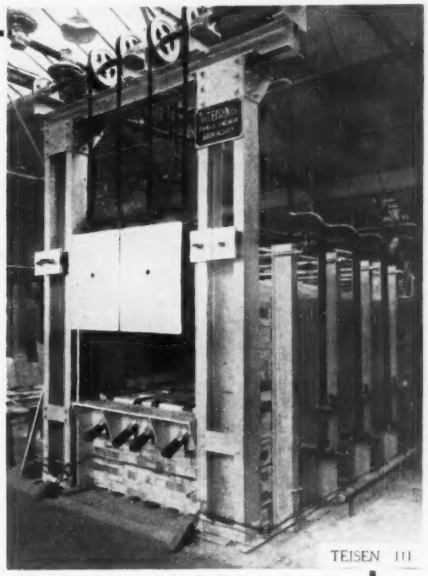

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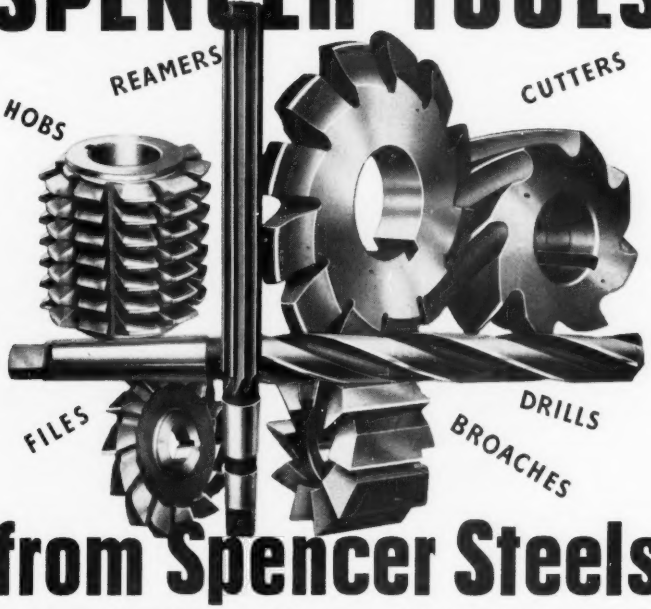

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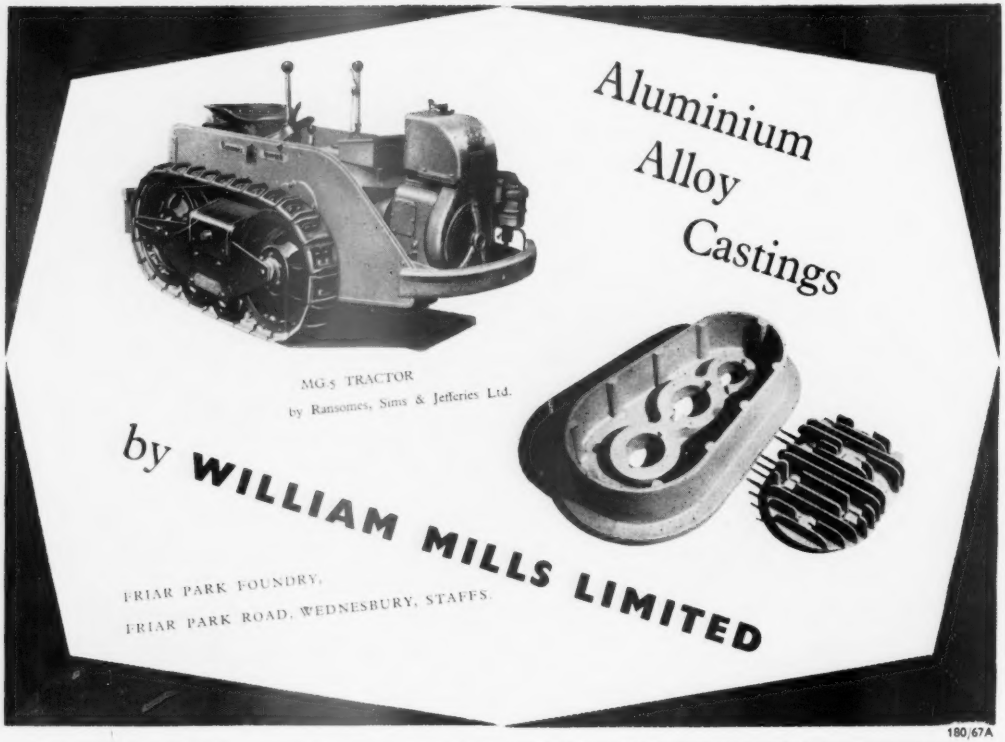

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INDEX TO MANUFACTURERS' ANNOUNCEMENTS

	PAGE
AC Sphinx Spark Plug Co.	84
Acheson Colloids, Ltd.	114
Acratork Engineering Co., Ltd.	128
Adamant Engineering Co., Ltd.	63
Aluminium Bronze Co., Ltd.	119
Aluminium Die Casting (Birmingham), Ltd.	110
Amal, Ltd.	110
Andrews, Harold, Grinding Co., Ltd.	95
Andre Rubber Co., Ltd.	68
Angus, George, & Co., Ltd.	91
Archdale, James & Co., Ltd.	42
Automotive Products Co., Ltd.	9, 10, 11, 12
Barber & Colman, Ltd.	127
B.B. Chemical Co., Ltd.	108
Benton & Stone, Ltd.	106
Birlec, Ltd.	36
Birmingham Aluminium Casting (1903) Co., Ltd.	Cover iv
Black & Decker, Ltd.	24
Booth, James & Co., Ltd.	73
Bound Brook Bearings (G.B.), Ltd.	40
British Electrical Development Association	2
British Iron & Steel Federation	106
British Oxygen Co., Ltd., The	92
British Timken, Ltd.	Cover i
British Wedge Wire Co., Ltd.	94
Broom & Wade, Ltd.	64
Brown, David, Tool Co., The	99
B.S.A. Tools, Ltd.	79
Burton, Griffiths & Co., Ltd.	79
Cape Asbestos Co., Ltd., The	52, 53
Carr Fastener Co., Ltd.	128
Cary, Wm. E., Ltd.	128
Cassell Cyanide	54
C.A.V., Ltd.	82
Churchill, Charles, & Co., Ltd.	131
Clancey, G., Ltd.	134
Clayton Crane & Hoist Co., Ltd., The	96
Clayton Dewandre Co., Ltd.	50
Clayton-Wright, Howard, Ltd.	123
Cleveland Petroleum Co., Ltd.	14
Climax Molybdenum Co. of Europe, Ltd.	43
Cocker Bros., Ltd.	87
Cohen, Geo., Sons & Co., Ltd.	119
Cole, E. K., Ltd.	132
Collier & Collier, Ltd.	118
Cooper & Co. (Birmingham), Ltd.	112
Coopers Mechanical Joints, Ltd.	18
Cossor, A. C., Ltd.	Cover iii
Coventry Motor Fittings Co., Ltd.	25
Dale, John, Ltd.	75
Darwins, Ltd.	100
Desoutter Bros., Ltd.	32
Dobbie, McInnes, Ltd.	122
Doncaster, Daniel, & Sons, Ltd.	111
Dualloys, Ltd.	109
Dunford & Elliott (Sheffield), Ltd.	112
Dunlop Rubber Co., Ltd.	93
Eagle Pencil Co., Ltd., The	108
Eaton Axles, Ltd.	51
Edwards, F. J., Ltd.	129
Electric Furnace Co., Ltd.	94
English Electric Co., Ltd.	59
English Steel Corporation, Ltd.	Cover ii
Feeny & Johnson, Ltd.	132
Feltham, Walter H., & Sons, Ltd.	136
Fenner, J. H. & Co., Ltd.	118
Ferodo, Ltd.	69
Fibreglass, Ltd.	41
Firth Brown Tools, Ltd.	5
Fletcher Miller, Ltd.	98
Fortis Tools, Ltd.	136
Fuller Gear Boxes	81
Fuller, Horsey, Sons & Cassell	104
Funditor, Ltd.	125
Gas Council, The	6
General Electric Co., Ltd.	8, 103
Girling, Ltd.	45
Glacier Metal Co., Ltd., The	105
Graham, H. G., & Son, Ltd.	120
Griffiths, Gilbart, Lloyd & Co., Ltd.	135
Groves, Samuel, & Co., Ltd.	132
G.W.B. Electric Furnaces, Ltd.	16
Habershon, J. J., & Sons, Ltd.	137
Hairlok Co., Ltd., The	110
Hardy, Spicer & Co., Ltd.	101
Harper, John & Co., Ltd.	57
Harris, John, Tools, Ltd.	49
Herbert, Alfred, Ltd.	31
Hey Engineering Co., Ltd.	100
Hoffmann Manufacturing Co., Ltd., The	61
Holroyd, John & Co., Ltd.	74
Humphreys, J. H. & Sons	124
Heywood, W. H., & Co., Ltd.	126
Hymatic Engineering Co., Ltd., The	115
I.C.I. (Marston Excelsior, Ltd.)	102
Imperial Smelting Corporation, Ltd.	113
Insley (London), Ltd.	136
International Twist Drill Co., Ltd., The	121
Iso-Speedic Co., Ltd., The	28
Jackson, H., Ltd.	21
Jenks Bros., Ltd.	122
Jenks, A. F. & Cattell, Ltd.	130
Jessop, William, & Sons, Ltd.	97
Johansson, C. E., Ltd.	92
Johnson & Johnson (Great Britain), Ltd.	112
Kendall & Gent, Ltd.	90
Kirkstall Forge Eng., Ltd.	33, 116
Klinger, Richard, Ltd.	62
Laycock Engineering Co., Ltd.	26
Lewis Spring Co., Ltd.	116
Leys Malleable Castings, Ltd.	37
Lindley, C., & Co., Ltd.	126
Linread, Ltd.	30
Marbaix, Gaston E., Ltd.	58
Marshall, C. & C., Ltd.	130
M.C.L. and Repetition, Ltd.	136
Mek-Elek Engineering, Ltd.	132
Metalastik, Ltd.	83
Midland Mechanical Developments, Ltd.	13
Midland Motor Cylinder Co., Ltd., The	23
Mills, William, Ltd.	138
Mond Nickel Co., Ltd., The	27
Monks & Crane, Ltd.	7
Monochrome, Ltd.	124
Morgan Crucible Co., Ltd., The	86
Moss Gear Co., Ltd.	96
Murad Developments, Ltd.	47
Nash & Thompson, Ltd.	123
Neill, James & Co. (Sheffield), Ltd.	125
Newall Group Sales, Ltd.	60
Newall, A. P. & Co., Ltd.	76
Nitralloy, Ltd.	134
Northern Automatic Screw Co., Ltd.	134
Perry Barr Metal Co., Ltd.	121
Plastomatic, Ltd.	107
Pollard Bearings, Ltd.	80
Ransome & Marles Bearing Co., Ltd.	56
Redifon, Ltd.	117
Redfern's Rubber Works, Ltd.	102
Renfrew Foundries, Ltd.	38
Renold & Coventry Chain Co., Ltd., The	15
Resinoid & Mica Products, Ltd.	114
Rockwell Machine Tool Co., Ltd.	89
Rubber Bonders, Ltd.	34
S. & D. Rivet Co.	120
Seal-Less Strapping, Ltd.	126
Sheffield Twist Drill & Steel Co., Ltd., The	71
Shell-Mex & B.P., Ltd.	20
Silentbloc, Ltd.	55
Skefko Ball Bearing Co., Ltd. The	88
Slick Brands, Ltd.	133
Small & Parkes, Ltd.	66
Smethwick Drop Forgings, Ltd.	72
Smith, W. H., & Son	114
Solus-Schall, Ltd.	122
South Wales Forgemasters, Ltd.	127
Spencer & Halstead Ltd.	131
Spencer, Walter & Co., Ltd.	137
Steel Company of Wales, Ltd., The	35
Stein & Atkinson, Ltd.	78
Sterling Metals, Ltd.	67
Tecalemit, Ltd.	19
Tenaplas (Sales), Ltd.	124
Terry, Herbert, & Sons, Ltd.	4
Teisen Th., C. E. Furnace Engineer	135
T.I. Aluminium, Ltd.	85
Tilghman's Patent Sand Blasting Co., Ltd.	98
Toledo Woodhead Springs, Ltd.	70
Tube Products, Ltd.	48
Udal, J. P., Ltd.	104
Vandervell Products, Ltd.	77
Vaughan, Edgar, & Co., Ltd.	17
Walker, M. W., & Staff, Ltd.	29
Ward, H. W. & Co., Ltd.	90
Ward, Thomas W., Ltd.	118, 133
West Yorkshire Foundries, Ltd.	39
Weston, Charles & Co., Ltd.	116
Whyte & Collins, Ltd.	130
Wickman, Ltd.	1, 3
Wiggin, Henry & Co., Ltd.	65
Wilmot-Breeden, Ltd.	22
Woodhead, Jonas, & Sons, Ltd.	46
Yarwood, Ingram & Co., Ltd.	129
Zinc Alloy Diecasters' Association	44

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